HANDBOOK

FOR

HEATING AND VENTILATING ENGINEERS

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FOURTH EDITION
REWRITTEN AND RESET

McGRAW-HILL BOOK COMPANY
239 WEST 39TH STREET, NEW YORK
6 BOUVERIE STREET, LONDON, E. C.
1920

First Edition, 1910
Second Edition, 1913
Third Edition, 1917
Fourth Edition, 1920

PREFACE TO FOURTH EDITION.

Changes in the art of heating and ventilating buildings have been so pronounced in the last few years that it has been considered advisable to entirely reconstruct the Handbook rather than to make additions to the old text. The book, therefore, has been rewritten and reset in every part. There have been added approximately 87 pages consisting of revisions, extended discussions of original text and new subject matter not before considered. Of this increase, Chapter I has 17 pages, including discussions on heat applications, combustion of fuels and analysis of flue gases; Chapters II, III, IV and V on air measurements, heat losses and furnace heating have 19 pages devoted largely to extensions; Chapters VI, VII, VIII and IX on hot water and steam heating have 28 pages, increasing the original text of this part by approximately 43 per cent. This includes descriptions of modified gravity systems, both steam and water, valves, fittings and piping connections; Chapters X, XI and XII on mechanical warm air systems have 10 pages of extensions, and Chapter XIII has 4 pages of extensions to the calculations of hot water and steam mains. The remainder of the book is in substance as it was with the addition of *Suggestions to School Districts,* 4 pages, Chapter XVIII, *Suggested Piping Connections for Vacuum System Details,* 3 pages, Appendix 3, and several new tables on pipe sizes for hot water and steam service.

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Especial attention has been given to the simplification of every important subject by applications to practical problems. These applications in most cases have been completely analyzed and their results compared with other parallel cases. No effort has been spared to have the entire subject matter complete and up to date and to present it in a way that will be at once simple and effective.

This little book is as a growing child. We wish it to be very active and useful to the general public. To do this it must be versatile and resourceful, carrying no excess material and trained down to service condition. We ask the assistance of our friends and their suggestions in its behalf.

LaFayette, Ind. J. D. H.

EXTRACT FROM PREFACE TO FIRST EDITION.

In the development of Heating and Ventilating work, it is highly desirable that those engaged in the design and the installation of the apparatus be provided with a Handbook convenient for pocket use. Such a treatise should cover the entire field of heating and ventilation in a simplified form and should contain such tables as are commonly used in every day practice. This book aims to fulfill such a need and is intended to supplement other more specialized works. Because of the scope of the work, its various phases could not be discussed exhaustively, but it is believed that the fundamental principles are stated and applied in such a way as to be easily understood. It is suggestive rather than digestive. The principles presented are the same as those that have been stated many times before, but the arrangement of the work, the applications and the designs are all original. Many equations and rules are necessarily given; but it will be seen that, in most cases, they are developments from a few general equations, all of which can be readily understood and remembered. Practical points in constructive design have also been considered. However, since the principles of heating and ventilation are founded upon fundamental thermodynamic laws, it seems best to accentuate the theoretical side of the work in the belief that if this is well understood, practical points of experience will easily follow.

All the standard works upon the subject have been freely consulted and used. In most cases where extracts are made, acknowledgment is given in the text. Because of these references throughout the book, we do not here repeat the names of their authors. We wish, however, to express our sincere appreciation of their valuable assistance.

LaFayette, Ind. J. D. H.

EXTRACT FROM PREFACE TO SECOND EDITION.

A few corrections were made on the first edition and all the material has been revised and brought up to date. The work on air conditioning has been amplified. The descriptions of hot water and steam heating have been improved by diagrams of the various piping systems. Two chapters have been added on refrigeration and many tables have been added in the Appendix. Many suggestions coming from men in practice have been included, thus enlarging upon the practical side and the applications.

Lincoln, Neb. J. D. H.

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CHAPTER I.

HEAT—ITS NATURE, GENERATION, USE, MEASUREMENT AND TRANSMISSION.

1. Introductory:—In all localities where the atmosphere drops in temperature much below 60° Fahrenheit, there is created a demand for the artificial heating of buildings. As the buildings have grown in size and complexity of construction, so also this demand has grown in extent and preciseness, with the general result that out of the open fire place and iron stove there has developed a science growing richer each day from inventive genius and manufacturing technique—the science of the heating and ventilating of buildings. The purpose of this handbook shall be to outline the fundamental principles and practical applications of this science in its various branches.

To the average heating engineer it may be that the exact nature of heat itself is of much less moment than its generation and transmission, but these facts should be impressed,—that heat is one form of molecular energy, that it cannot be created except by conversion from some other form, and that it is infallibly obedient to certain physical laws and principles which should be understood and used by every engineer.

In generating heat for heating purposes the almost universal method is combustion. The union of the combustible content of such substances as coal, wood or peat with the oxygen of the air is always attended by a liberation of heat derived from the chemical action of the combination; and this heat is carried by some common carrier, such as air, water or steam, to the building or room to be heated where it is given off by the natural cooling process. In some instances this heat is converted into electrical energy which is carried by wire to the place of use and given off as heat through a set of resistance coils. This method is not much favored as yet because of its inefficiency and the resulting expense, an objection which does not hold in the case of water power installations where the combustion of fuel is entirely eliminated.

2. Heat—Temperature:—The meaning of the word *heat* should not be confused with that of the word *temperature.* Although closely related they are far from being interchangeable. In a given mass of any substance, except when passing through a change of state, the universal law is that the addition of heat raises the temperature and the subtraction of heat lowers the temperature of the substance. Heat is the cause and temperature is one of the effects. In the measurement of heat the most commonly accepted unit in practical engineering work is the *British thermal unit,* abbreviated B. t. u. This may be defined as *that amount of heat which will raise the temperature of one pound of pure water one degree Fahrenheit* (See definition for specific heat, Art. 8). This unit value, the B. t. u., measures the quantity of heat, while the temperature measures the intensity or degree of heat. In equal masses of the same substance the two are proportional. The *Fahrenheit* scale is the more commonly used temperature scale, especially in steam engineering. The unit of this scale is derived by dividing the distance on the thermometer between the freezing point and the boiling point of water into 180 spaces called degrees, the freezing point being marked 32° and the boiling point 212°. *All temperatures in this book, unless otherwise stated, will be taken according to the Fahrenheit scale and all quantities of heat expressed in British thermal units.*

A second unit of quantity of heat considerably used in scientific research is the *calorie,* abbreviated cal., and defined as that amount of heat which will raise one kilogram of pure water from 17° to 18° centigrade. The *centigrade* scale is a second temperature scale, the unit of which is derived by dividing the distance on the thermometer between the freezing point and the boiling point of water into 100 degrees, the freezing point being marked 0° and the boiling point 100°.

It is often found desirable to change the expression for temperature or for quantity of heat from one system of terms to that of the other. For this purpose the following equations will be found useful:

$$F = \frac{9}{5} C + 32 \text{ and } C = (F - 32) \frac{5}{9} \tag{1}$$

where F = Fahrenheit degrees and C = centigrade degrees. From these equations it may be seen that the two scales

coincide at but one point, — 40°. For conversion of the quantity units the following may be used:

1 British thermal unit = 0.252 calorie.

1 calorie = 3.968 British thermal units.

These are for the absolute conversion of a certain quantity of heat from one system to the other. If, however, the *effect* of this heat is considered upon a given weight of substance and the weight also is expressed in the respective systems, the following values hold:

1 calorie per kilogram = 1.8 British thermal units per pound.

1 British thermal unit per pound = 0.555 calorie per kilogram.

For conversion tables see Marks' Mechanical Engineers' Handbook or Kent's Mechanical Engineers' Pocket-Book.

3. Instruments Used in Measuring Temperature:—Instruments intended to indicate degree or intensity of heat, i. e., the temperature of substances, are designed upon many different principles. Of these the following represent the important general classifications:

Expansion of a Liquid with Increase in Temperature.—The ordinary mercury, alcohol or ether-in-glass *thermometers* belong to this great class. Mercury thermometers should not be used to register temperatures near the top of the scale for fear of rupturing the glass. To overcome this difficulty some thermometers are made with a mercury-well at the upper end of the mercury column. The objection to be offered to this form is the difficulty of completely emptying the upper well after it has been partially or wholly filled with mercury. The ordinary mercury-in-glass thermometer, either with or without the upper mercury-well, should not be used on temperatures above 600° F. because of the fact that mercury boils at 680° F. Mercury-in-glass or mercury-in-quartz thermometers have been used up to 1300° F. by compressing into the space above the mercury some neutral gas, as nitrogen or carbon dioxide. This type, however, is open to the objection of high breakage costs. Due to the fact that mercury freezes at —38° F. it cannot be used for low temperature thermometers. These are usually made with alcohol as the liquid, since alcohol freezes at —170° F.

Expansion of a Solid with Increase in Temperature.—Instruments built upon this principle are commonly called *expansion pyrometers*. Fig. 1, *a*, shows such a pyrometer. Inside

the stem of the instrument is a metallic expansion element, the movement of the free end of which operates the hand on the dial. Such an instrument may be used up to the lowest temperature of the softening point of the metals in the stem. Ordinarily, errors of 2 to 5 per cent. may be expected in the temperature reading.

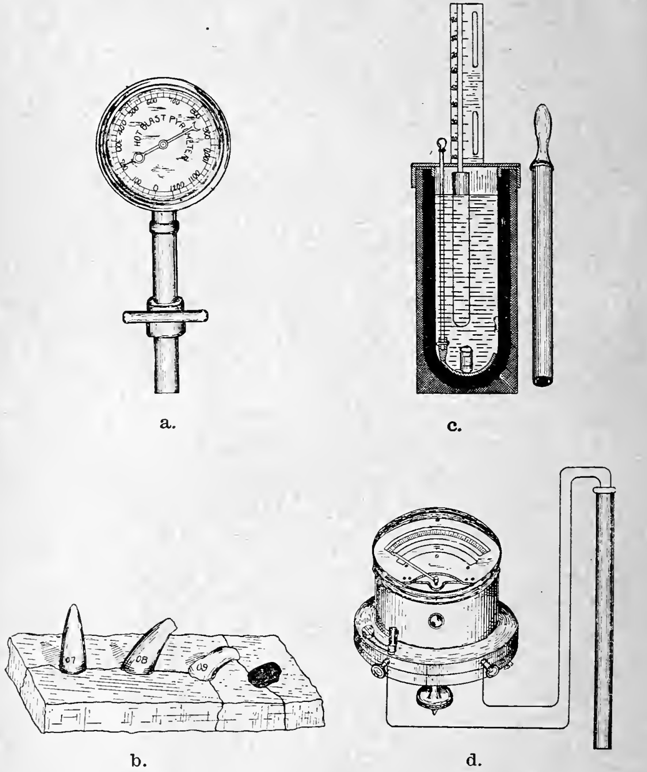


Fig. 1.

Fusion of Cones of Refractory Materials.—This principle is exceedingly simple in application as shown in Fig. 1, *b*. Several of a series of cones, varying in mineral compositions and hence in melting points, are exposed to the temperature

to be measured and this temperature is indicated by that cone of the series which just melts or softens sufficiently to lose its shape. With the cones is furnished a table of temperatures for comparison. From the illustration, the temperature indicated is evidently that corresponding to cone number 08, which from the Seger cone table is 1814° F. *Seger cones* for such measurements may be obtained to indicate temperatures from 1094° F. to 2800° F. by increments varying from 25 to 55 degrees.

TRANSFER OF A HIGH TEMPERATURE BODY AND ITS HEAT TO A KNOWN QUANTITY OF WATER.—This is the principle embodied in all *pyrometers* of the *calorimetric* type, one of which is shown in Fig. 1, *c*. A thoroughly insulated vessel contains a known quantity of water, a thermometer and a stirring device. A ball of platinum, copper or iron of known weight and specific heat is exposed to the temperature to be measured, by means of the handle shown in the figure or by a small crucible. When the ball has reached its upper temperature, it is quickly transferred to the water of the insulated vessel and the rise of temperature of the water is noted from the thermometer. Upon the suppositions that all the heat in the ball is transferred to the water and that the ball and the water finally reach the same temperature, the assumption may be made that the heat gained by the water equals that lost by the ball, hence the product of the weight, temperature rise and specific heat of the water, divided by the product of the weight and specific heat of the ball gives the drop in temperature through which the ball has passed. From this the upper temperature reached by the ball may be obtained by adding to the temperature drop, the final temperature of the water and the ball. Let s = specific heat of the ball, T = upper temperature of ball, m = weight of the ball, t and t' respectively = beginning and ending temperatures of the water, and w = weight of the water. Remembering that the specific heat of water is 1, we have

$$w\ (t' - t) = sm\ (T - t') \text{ whence} \tag{2}$$

$$T = \frac{w\ (t' - t)}{sm} + t'$$

The objections to this method of temperature measurement are its slowness due to the necessary computations and manipulations, and the fact that considerable error may be introduced during the transference of the ball from the heated space to the calorimeter. When this method is used

for very high temperatures the ball is made of porcelain or fire clay.

Change of Resistance of an Electric Conductor, or Change of Voltage of an Electric Thermo-couple.—Instruments built upon either of these two electrical principles are extremely delicate but give very accurate results, it being possible to determine temperatures up to 2000° F. with a variation of but one or two degrees. For practical work *electric pyrometers* are more commonly of the *thermo-couple* type (See Fig. 1, *d*). To the right is shown a porcelain tube enclosing a thermo-couple of two dissimilar metals. If this tube is subjected to the temperature to be measured, the potential generated by the couple upon heating is proportional to the temperature. Hence, if connected to a voltmeter as shown at the left, the voltage generated may be indicated, or as is usual, the temperature may be read directly since the scale of the voltmeter may be graduated in degrees instead of in volts. This type of pyrometer is extensively used. From each of a large number of testing points, thermo-couple wires may be brought to a central point, where by means of a switch the temperature at any couple may be instantly observed by throwing its current into a common voltmeter or temperature indicator.

Other types of temperature measuring instruments are designed upon the principle of the optical pyrometer, and the gas and air thermometers, but these are not used to as large an extent in practice as are the five above mentioned.

4. Absolute Temperature:—In experiments that have been carried on with pure gases with the use of air thermometers, it has been found that gases expand or contract approximately $\frac{1}{492}$ of their volumes at 32° F. ($\frac{1}{460}$ of their volumes at zero F.) per degree change in temperature, or $\frac{1}{273}$ of their volumes at zero C. From the same line of reasoning, by cooling a gas to —460° F. or —273° C, it would cease to exist. This theoretical point is called the absolute zero of temperature. All gases change to liquids or solids before this point is reached, however, and hence do not obey the law of contraction of gases at very low temperatures. The fact that air at constant pressure changes its volume almost exactly in proportion to the absolute temperature, T, ($460 + t$, where t is temperature Fahr-

enheit) gives a starting point to be used as a basis for all air volume-temperature calculations. For instance, if a volume of 20000 cubic feet of air at 0° is heated to 70° with constant pressure, its volume after heating will be greater in the same proportion as its absolute temperature is greater; that is, $\frac{x}{20000} = \frac{530}{460}$; $x = 23000$ cubic feet, or an increase of 15 per cent.

5. Gage and Absolute Pressures:—Gage pressure is the total pressure per square inch in a container minus the pressure of one atmosphere. Thus 65 pounds gage pressure means that the container is carrying 65 pounds pressure per square inch of surface above the pressure of the atmosphere. Atmospheric pressure at sea level, 14.696 commonly written 14.7, is used on all but the most exact calculations. This pressure becomes less as the elevation rises above sea level. As a general statement it may be said that atmospheric pressure reduces ½ pound for each 1,000 feet above sea level (See Table 8, Appendix). The total pressure exerted within the container is therefore $65 + 14.696 = 79.696$ at sea level. This total pressure is known as the *absolute pressure* and when stated in pounds per square foot of area is called *specific pressure.*

6. Mechanical Equivalent of Heat:—By experiment it has been determined that if the heat energy represented by one B. t. u. be changed into mechanical energy without loss, it would accomplish 778 foot pounds of work. Similarly, one calorie is equivalent to 428 kilogrammeters of work.

7. Latent Heat, Total Heat, Etc.:—Not all the heat applied to a body produces change in temperature. Under certain conditions the heat applied produces internal or molecular changes and is called *latent heat.* Thus in a normal atmosphere if heat is applied to ice at the freezing point, no rise of temperature is noted until all the ice is melted; and again, heat applied to water at the boiling point does not raise its temperature until all the water is changed to steam. The first is called latent heat of fusion, which for ice is 144 B. t. u. per pound; the latter is called latent heat of vaporization, which for water is 970.4 (Marks and Davis) B. t. u. per pound. For most calculations the approximate value 970 may be used. Consult books on thermodynamics for further discussion of latent heat as composed of internal and external work equivalents. *Sensible heat*

is that heat whose addition or subtraction can be detected by a thermometer. As applied to the standard steam tables, this is equal to the total heat above 32° minus the latent heat of vaporization. *Heat of the liquid,* as applied to the standard steam tables, is that quantity of heat added to a pound of water at 32° to bring it to the temperature of the boiling point at any given pressure. At atmospheric pressure this is 180 B. t. u. *Total heat* is that quantity of heat represented by the sum of the latent heat of vaporization and the heat of the liquid. In the evaporation of water at atmospheric pressure this is $970.4 + (212 - 32) = 1150.4$ B. t. u. Total heat is different for all pressures at which evaporation takes place. Consult Art. 14 and Table 4, Appendix, for latent heat, heat of the liquid and total heat at different pressures.

Convenient approximate equations for latent heat and total heat are those quoted by Regnault.

$$\textit{Latent Heat} = 1092 - .695\,(t - 32) \qquad (3)$$

where t = temperature at which the steam is formed.

Illustration.—The latent heat of steam at a temperature of 338° (pressure 100 lbs. gage) is $1092 - .695\,(338 - 32) = 879.3$ B. t. u.

$$\textit{Total heat} = 1092 + .305\,(t - 32) \qquad (4)$$

Illustration.—The total heat above 32° of the same steam as in previous illustration is $1092 + .305\,(338 - 32) = 1185.3$.

8. Specific Heat:—The specific heat of a substance is that quantity of heat added to or subtracted from a unit weight of the substance when its temperature is changed one degree. The *mean specific heat* is that quantity of heat added to or subtracted from a unit weight of the substance in changing through any given number of degrees, divided by the number of degrees change. For illustration, the specific heat of water that has been considered standard for many years is obtained at the temperature of its maximum density, 39.1° F. (4° C.). This is used in much of the physical and scientific calculations, but in most engineering work the tendency is to take the mean specific heat between the temperatures of 32° F. and 212° F. (0° C. and 100° C.), i. e., the heat required to raise one pound of pure water from 32° F. to 212° F. divided by 180. This is the same as the specific heat of water at 62° F. and agrees with the accepted value of the B. t. u. Table 26, Appendix, gives specific heats of substances.

9. **Radiation:**—Heat may be transmitted as a wave motion in the ether of space. In this way the heat of the sun reaches the earth. Heat of this form, usually referred to as radiant heat, requires no matter for its conveyance; passes through some materials, notably rock salt, without change or appreciable loss; and follows the laws for the radiation of light. It is assumed that the heat received by the atmosphere is obtained through contact with the bodies giving and receiving heat and that little is obtained directly from the radiant ray.

TABLE 1.
Radiation Constants, Values of C

Material	C
Glass, smooth	0.154
Brass, dull	0.0362
Copper, slightly polished	0.0278
Lampblack	0.154
Wrought-iron, dull, oxidized	0.154
Wrought-iron, clean, bright	0.0562
Cast iron, rough, highly oxidized	0.157
Lime plaster, rough, white	0.151
Slate	0.115
Gold plate, shining but not polished	0.082
Clay	0.065

The capacity that any body has of absorbing the radiant ray is called its *absorption capacity.* Absolute black bodies theoretically absorb all the radiation received upon their surfaces and have an absorption capacity of 1. Bright or polished surfaces have a reduced absorption capacity. It is also understood that the *radiation capacity* is proportional to the absorption capacity. The *amount of heat radiated* by a substance is practically independent of the form of the surface and depends upon the difference of temperature between the radiating and receiving surfaces, and upon the color and character of the surfaces. The Stefan-Boltzman radiation law states that for black bodies the radiating power is proportional to the fourth power of the absolute temperature of the body. For other than black bodies this law is also approximately true. Let R = area of radiating surface in square feet, H = B. t. u. radiated per hour, T =

absolute temperature of the substance, and C = a constant; then, $H = CR\ (T \div 100)^4$. For a dead black body C = .1618. Other values of C from Hutte are shown in Table 1.

Assuming in general that radiating surfaces for heating systems may be classified as black bodies, the amount of heat radiated from a surface R having an absolute temperature T to surrounding surfaces having an absolute temperature T_1 is

$$H = C\ R\ [(T \div 100)^4 - (T_1 \div 100)^4]$$

Applications of the theoretical formula of radiant heat to practical problems in general give very unsatisfactory results.

10. Conduction:—This method of heat transmission is very evident to the senses. If a rod of metal is heated at one end, the heat is transferred or conducted along the rod by molecular action. Conduction being essentially the way by which solids transfer heat, it is of special significance in the calculation of heat losses through the walls of a building. The *coefficient of conduction* may be defined as that quantity of heat which passes through a unit thickness of substance in a unit of time across a unit of surface, the difference of temperature between the two sides of the substance being one unit of the thermometric scale employed. The amount of heat conducted through a material in a given time is directly proportional to the difference in temperature between the two parallel sides of the substance and inversely proportional to the thickness. As a formula $H = c/b\ (t_1 - t_2)$ where c = coefficient of conductivity, b = thickness of material in inches, and t_1 and t_2 = respective temperatures. Since the complexity of building constructions renders it impossible to reduce all conduction losses to losses per unit thickness of the structure, the term *rate of transmission* may be used instead of conductivity and may be understood to include combinations of conductivities and thicknesses. This may be illustrated by the ordinary framed and studded wall where K is the rate for the combination (See Chapter III).

11. Convection:—Gases and liquids convey heat most readily by this method, which is fundamental with warm air and hot water heating installations. If it is attempted to heat a body of water by applying heat to its upper surface, it will be found to warm up with extreme slowness. If, however,

the source of heat be applied below the body of water, it will be found to heat rapidly. What actually happens is this: water particles near the source of heat become lighter, volume for volume, than the colder particles near the top, and because of the change in density gravity causes an exchange of these particles, drawing the heavier to the bottom and allowing the heated and lighter particles to rise to the top thus forming circulation currents. This process is known as *convection.* It will not occur unless the medium expands upon being heated and unless the force of gravity is free to establish circulating currents. In the hot water heating system (Fig. 2), water rises by convection to the radiators, is there cooled and descends by the return circuit to the point of heat application completing the circuit. The warm air furnace installation works similarly, air, however, being the heat-carrying medium.

Fig. 2.

12. Work:—*Work is the overcoming of a resistance along a line of motion.* It is the product of force and distance and is independent of time. Assuming the pound to be the unit of force and the foot to be the unit of distance, the unit of work is the *foot-pound.* To lift one hundred pounds one foot or one pound one hundred feet would cause the expenditure of one hundred foot pounds of work.

13. Power:—Power and work are closely related but are not identical. *Power is the rate of doing work* and always comprehends the element of time. The unit of power, called *horse-power,* has no reference to the power of the horse nor to the boiler horse-power, but is an arbitrary value equivalent to

1 horse-power =
746 Watts = .746 K. W.
33000 ft. lbs. of work per min.
4562.4 kilogrammeters of work per min.
$33000 \div 778 = 42.416$ B. t. u. per min.
$4562.4 \div 428 = 10.66$ cal. per min.

If 100 cubic feet of water, weighing 62.5 pounds per cubic foot, are lifted 100 feet per minute without friction loss, the horse-power is $(100 \times 62.5 \times 100) \div 33000 = 18.94$.

The term *boiler horse-power* is equivalent to 34.5 pounds

of water per hour evaporated from water at 212° F. to steam at 212° F. This equals $970.4 \times 34.5 = 33479$ B. t. u.

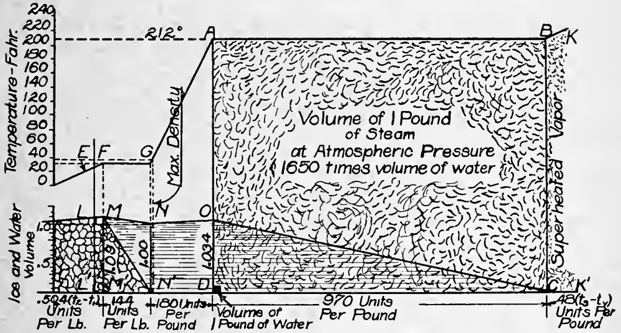
14. Application of Heat to Solids and Liquids:—All matter in its most finely divided state is made up of minute particles called atoms which are drawn together by a force called attraction. This attraction is lessened by the application of heat, the particles tending to separate (substance increasing in size) until such a temperature is reached (a certain amount of heat is absorbed) when the attraction is zero. From this point further application of heat will cause repulsion and the particles will fly apart. This explains the existence of the three states of matter; solid, liquid and gaseous. No two substances act exactly alike upon the addition or subtraction of heat, but practically all substances under certain conditions may exist in any one of the three states. The exact points of separation between the solid, liquid and gas, differ very much in different substances; but regardless of this fact, each substance no matter what its state may be solidified by cooling or vaporized by heating. The amount of heat that may be carried by any substance in any given state is called its *capacity for heat.*

When *solids* change in temperature they change in volume in practically all cases, increasing with rise of temperature and decreasing with fall of temperature. This fact many times causes considerable annoyance to any one manufacturing or using materials of construction. Since all metals that enter into engineering construction are subject to sudden and sometimes very extreme changes of temperature, it is frequently necessary to put in compensating devices to account for such temperature changes. The steel framework of a building for example is subjected to extremes of summer and winter temperatures, causing change in the building size. This change is small, but during the cold weather when the building materials have a slight reduction in size, the steam pipes are under high temperatures and have their maximum size. During the summer when no heat is necessary, reverse conditions exist. In high buildings this change is sufficient to demand compensators or expansion joints in the steam lines, otherwise there would be contact between the pipes and the building which might be sufficient to rupture some part. Like conditions exist in street mains (conduit lines), basement mains in buildings, horizontal connections between vertical risers,

riser connections between floors, boiler pipe connections, boiler settings in brick work and in many other places around the heating system of the average building.

Sudden changes of temperature in any material are to be avoided when possible. This is especially true if the materials are fastened together with screws, bolts or rivets, such as boilers, heaters or piping systems. When heat is thus applied it is always more intense at one place than at another and the expansion or contraction is not uniform, causing unnecessary stresses and many times leaks and ruptures. The *force exerted by heat* in expanding any substance is the same as would be required to stretch the same substance an equal amount by mechanical means or to compress the enlarged piece to its former size.

When heat is applied to *liquids*, the phenomenon of expansion is apparent as in solids. One notable exception is found in water between 32° and 39.1° F. as will be seen later. *Since water is the liquid universally used in heating systems,*



VOLUME—TEMPERATURE—STATE CHANGE OF WATER UNDER ATMOSPHERIC PRESSURE.

Fig. 3.

it is of interest to study its characteristics under different conditions of heat. Start with a mass of one pound of ice at some temperature, say 25° F. (it must be remembered that after ice is formed at 32° F. it may be cooled to any temperature below 32° by the continued extraction of heat), and while heat is being added to the mass, note the changes taking place. In Fig. 3 *EFGABK* is the *temperature curve, LMNOC* is the *volume curve,* the ordinates *MM′, NN′, OD* and *BC* represent

the *periods of change of state,* and the horizontal line at the base of the chart $L'C$ represents *heat units added.* With LL' representing the volume of a pound of ice at any temperature (in this case 25° F.) heat is added and the temperature curve E rises to F. The quantity of heat added is found by multiplying the pounds of ice (in this case 1) by the specific heat, which for ice is .504, and by the rise in temperature. The addition of .504 B. t. u. for each degree rise between 25° and 32° gives the ice $(32 - 25) \times .504 = 3.528$ B. t. u. and brings it to the temperature of the melting point. While the temperature has been gradually increasing the volume has also increased slightly. See MM'. More heat is added and the ice begins to melt but the temperature does not rise as would be expected. It remains constant from F to G until all the ice has changed to water, as shown by the line MN'. In this change there has been a reduction in the volume of the mass as shown by the dropping of the line MN. Notice that the volume of the water is taken as 1 and the volume of the ice at 32° as 1.09. This explains why water allowed to freeze in a pipe often causes the bursting of the pipe. The quantity of heat absorbed during the change of state from ice to water without change of temperature is found by experiment to be 144 B. t. u. per pound and is called the latent heat of fusion. Conversely, in the reverse change the same amount of heat would be given off. So far we have added to the pound of ice $3.528 + 144 = 147.528$ B. t. u. and have increased the temperature only 7 degrees. From this point GNN', where the entire mass is water with a volume approximately equal to 1, the addition of heat causes a uniform rise in temperature along GA; also a slight decrease in volume along MN to the point of maximum density 39.1°, where the volume NN' is 1, and from here a uniform increase in volume along NO until the temperature has risen from 39.1° to 212° and the volume has increased from 1 to 1.034, with an addition of $212 - 32 = 180$ B. t. u. To arrive at the state line AOD required the addition of $3.528 + 144 + 180 = 327.528$ B. t. u., and a total of 187 degrees change. At AOD a second change of state is encountered. 970.4 B. t. u. (latent heat of vaporization) are now added to the pound of water without changing its temperature and the mass has a uniform change of state from water at 212° to steam at 212°. When the temperature line reaches B the volume line of the water is at C, indicating that all the

water has become steam at atmospheric pressure and now occupies a volume *DABC*, 1650 times the volume of the water that produced it (compare volume *ABCD* with small black volume *D*). The pound of ice has now received 327.528 + 970.4 = 1297.928 B. t. u. and is in a state of steam at atmospheric pressure and 212° temperature. Any further addition of heat to this steam without being in contact with water results in an increase of temperature along the line *BK* and the steam is said to be *superheated.* The quantity of heat added as superheat is found by multiplying the pounds of steam (in this case 1) by the specific heat and by the change in temperature. For steam the specific heat varies with the pressure. A fair average value is .48. The heat absorbed for any degree of superheat may be added to the 1297.928 B. t. u. thus giving the *total heat* between the two extremes of temperature and pressure selected. Ordinarily heating calculations refer only to *saturated steam*, i. e., steam in contact with water and superheating need not be considered.

By the use of Equations 3 and 4 and the steam tables compare results by filling in the blank table the values for steam at 10, 14.7, 50 and 100 pounds absolute pressure.

	Equation				Table			
	10	14.7	50	100	10	14.7	50	100
Heat of the Liquid								
Latent Heat								
Total Heat								

Three standard tables of *properties of saturated steam* are in general use, Marks and Davis, Peabody, and Goodenough. These tables check each other closely and any one may be recommended (Table 4, Appendix, is an extract from the first table).

The following *summary of directions for the use of any of the steam tables* gives specific equations for the solution of almost any type of problem using any vapor table. With the nomenclature of Marks and Davis, we have:

FOR SUMMATION ABOVE 32° F.

	If Quality is 100%	If Quality is $X\%$	If Superheat is D degrees
Total Heat of Formation....	$H = h + L$	$h + xL$	$H + CpD$
Intrinsic Heat of Formation	$h + I$	$h + xI$	$h + I + CpD - (Apu)_s$
External Work of Formation	$(Apu)_v$	$(xApu)_v$	$(Apu)_v + (Apu)_s$

FOR SUMMATION ABOVE SOME FEED TEMPERATURE $= t$

	If Quality is 100%	If Quality is $X\%$	If Superheat is D degrees
Total Heat of Formation	$H - ht$ or $h + L - ht$	$h + xL - ht$	$h + L + CpD - ht$
Intrinsic Heat of Formation	$h + I - ht$	$h + xI - ht$	$h + I + CpD - (Apu)_s - ht$
External Work of Formation	$(Apu)_v$	$(xApu)_v$	$(Apu)_v + (Apu)_s$

In these tables the subscript v refers to the condition of non-superheats, while the subscript s refers to the condition of superheat. In the term Apu, the value of A is 1/778, $p =$ pressure in pounds per sq. foot and u is the increase in volume in cubic feet undergone during the process in question. Some vapor tables, (notably Peabody's) contain columns of Apu worked out and tabulated while with the use of other tables it is necessary to calculate the values of the Apu terms.

These tables emphasize those facts the neglect of which causes perhaps 90 per cent. of all steam table calculation errors, viz:

x *cannot* affect, *as a factor* any steam table value except L, I, and $(Apu)_v$.

The vapor tables are summations above 32° F., and for heat summations above any other temperature, *correction must be made.*

The external work available during formation is independent of the feed temperature.

15. Application of Heat to Gases:—Pressure-volume-temperature changes in gases may be found from *ideal* laws which apply with close approximation, or from *actual* laws (modifications of the ideal laws) designed to fit actual conditions. The ideal laws are much more easily applied and

give results that are close to average practice; consequently, they are used in most engineering calculations. Ideal laws are known as (1) The Law of Boyle or of Mariotte, (2) The Law of Charles or of Gay Lussac.

BOYLE'S LAW.—*When the temperature of a given weight of gas is maintained constant, the volume and the pressure vary inversely.* In many pressure-volume applications to gases the temperature change is either zero or so small as to be of no serious moment. This law applies in such cases. Let P, P_1, P_2, etc., = absolute pressures in pounds per square foot, and V, V_1, V_2, etc., = volumes in cubic feet at the respective pressures, then

$$PV = P_1 V_1 = P_2 V_2, \text{ etc.} \tag{5}$$

In other words, at a constant temperature the product of any pressure with its respective volume is a constant quantity. Thus if 100 cubic feet of air at 14.7 lbs. absolute pressure be changed to 50 cubic feet without change of temperature, the pressure will be $(14.7 \times 100) \div 50 = 29.4$ lbs. absolute, or 14.7 lbs. gage.

CHARLES' LAW.—When gases are heated, they react according to the Law of Charles; i. e., *the volume of a perfect gas at constant pressure,* or *the pressure of a perfect gas at constant volume, is proportional to its absolute temperature.* As before let P = absolute pressure in pounds per square foot, V = volume in cubic feet, and T = absolute temperature, then

$$\frac{PV}{T} = \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}, \text{ etc.} \tag{6}$$

Referring to the first part of the definition of this law, let the temperature of a cubic foot of gas (take air for illustration at atmospheric pressure) be 32° F., if $T = 32 + 460 = 492$, $P = 14.7 \times 144 = 2116.8$ and $V = 1$, then $\frac{PV}{T} = \frac{2116.8 \times 1}{492}$. Now if the temperature of the air is changed to some other temperature T_1, say 100° F. at the same pressure, $\frac{PV}{T} = \frac{P_1 V_1}{T_1}$ and, since $P_1 = P$, the new volume is

$$V_1 = \frac{2116.8 \times 1}{492} \times \frac{560}{2116.8} = \frac{560}{492} = 1.14\ V$$

Referring to the second part of the definition of the same law, take a cubic foot of air at atmospheric pressure and 32° F. and change its temperature to 100° F. while the vol-

ume remains constant at one cubic foot. Now, the pressures at constant volume are proportional to the absolute temperatures and

$$\frac{P \times 1}{492} = \frac{P_1 \times 1}{560}$$

$$P_1 = 1.14\ P, \text{ specific pressure}$$

$$p_1 = 1.14\ p, \text{ pounds per square inch.}$$

General Equation.—The volume occupied by a pound of air at any given pressure and temperature (specific volume) is the reciprocal of its density at that temperature. At 32° F. and atmospheric pressure this is $1 \div .0807 = 12.391$. Substituting $T_1 = (32 + 460)$, $P_1 = (14.7 \times 144)$ and $V_1 = 12.391$, in Equation 6 and reducing

$$PV = 53.3\ T \tag{7}$$

This is usually written $PV = RT$, where R is a constant which varies for different gases. In further study of this question, it is found that R represents the foot pounds of external work done when the temperature of one pound of gas is raised one degree at constant pressure. For air, as found above, it is 53.3. Having the value R for any gas and any two of the values P, V, or T, the third may be found. Note: in Equation 7 P and V must be specific pressure and volume, respectively. To illustrate, the pressure of one pound of air having a volume of 5 cubic feet and temperature of 100° F. is $P = (53.3 \times 560) \div 5 = 5969.6$ pounds specific pressure, or 41.5 per square inch absolute. Also, the volume of a pound of air having a pressure of 50 pounds per square inch absolute and a temperature of 60° is $V = (53.3 \times 520) \div (64.7 \times 144) = 2.97$ cubic feet.

16. Combustion of Fuels:—Fuels used for heat production are solid, liquid and gaseous, and contain carbon (C), hydrogen (H), oxygen (O), nitrogen (N), sulphur (S), and small amounts of water and ash. In combustion the most valuable of all of these constituents are carbon and hydrogen. Fuels with high percentages of carbon and hydrogen (heat producing agents) and low percentages of ash and water are the most desirable. Coal is the universal fuel, although oil and gas are frequently used. Carbon burns to carbon dioxide (CO_2) if supplied with sufficient air during combustion or to carbon monoxide (CO) if the air supply is restricted. The greatest economy is found when CO_2 is

produced. Hydrogen burns, forming water, and sulphur burns to sulphur dioxide (SO_2). Oxygen in the fuel has the same effect as the oxygen of the air in supporting combustion. Nitrogen has no appreciable chemical action during combustion, but it absorbs heat and is thrown away, hence it tends to reduce the efficiency of the furnace. Water in the fuel has little chemical effect. It absorbs heat in being evaporated and superheated and passes off with the gases, causing small loss. One pound each of the above elements of the coal when completely consumed gives off heat units as follows: C to CO_2 = 14600, C to CO = 4450, CO to CO_2 = 10150, H to H_2O = 62000 (frequently used 52000 to account for loss by evaporation and superheating), and S to SO_2 = 4000.

As an illustration of the chemical changes taking place in a furnace when a fuel is raised in temperature sufficiently high that the combustible unites with the oxygen of the air and produces combustion, burn completely one pound of coal containing $C = .78$, $H = .04$, $O = .03$, $N = .02$, $S = .02$, $H_2O = .01$, and ash $= .10$, and note the following points of interest:

(A) Theoretical total heat of the fuel by equation.

(B) Amount of air needed for complete combustion.
(a) By analysis. (b) By equation.

(C) Probable amount of air used for combustion.

(D) Temperature of the furnace when only the theoretical amount of air is used for complete combustion.

(E) Temperature of the furnace when the probable amount of air is passed through the furnace.

(F) Efficiency of the furnace.

Theoretical Total Heat of the Fuel (A).—From the heat values given the following theoretical equation (Du Long's formula) has been compiled:

$$\text{Total Heat} = 14600\ C + 52000\ (H - \frac{O}{8}) + 4000\ S \qquad (8)$$

and when applied to the coal sample as stated gives

$$\text{Total Heat} = 14600 \times .78 + 52000 \times (.04 - \frac{.03}{8}) + 4000 \times .02 = 13353 \text{ B. t. u.}$$

Equation 8 is used when the chemical composition of the fuel is known. When this is not known, the total heat is found in the laboratory by the use of calorimeters.

In most furnaces combustion is not perfect. Part of the carbon is burned to CO_2 giving off 14600 B. t. u. per pound and part to CO giving off 4450 B. t. u. per pound. To find the heat value of the coal in such cases use a modification of Equation 8.

$$\text{Heat liberated} = 14600\ C_1 + 4450\ C_2 + 52000 \left(H - \frac{O}{8}\right) + 4000\ S \qquad (9)$$

where C_1 and C_2 = weights of carbon per pound of coal burned to CO_2 and CO respectively. Suppose, for illustration, that the carbon goes half and half to CO_2 and CO, then the heat liberated is $14600 \times .39 + 4450 \times .39 + 52000 \left(.04 - \frac{.03}{8}\right) + 4000 \times .02 = 9395$. Compare this with the value obtained by Equation 8.

THEORETICAL AMOUNT OF AIR NEEDED FOR COMPLETE COMBUSTION (B).—

(a) Since the atomic weights (relative weights of unit volumes referred to $H = 1$) of $C = 12$, $H = 1$, $O = 16$, $N = 14$, and $S = 32$, we have

12 parts C unite with 32 parts O. (1 lb. C + 2.66 lbs. O = 3.66 lbs. CO_2)

12 parts C unite with 16 parts O. (1 lb. C + 1.33 lbs. O = 2.33 lbs. C)

2 parts H unite with 16 parts O. (1 lb. H + 8.00 lbs. O = 9.00 lbs. H_2O)

32 parts S unite with 32 parts O. (1 lb. S + 1.00 lbs. O = 2.00 lbs. SO_2)

from which may be found the oxygen required to unite with each element for complete combustion. From the coal analysis,

$.78 \times 2.66 =$	2.075	lbs. O for the carbon
$.04 \times 8.00 =$	.320	lbs. O for the hydrogen
$.02 \times 1.00 =$	.020	lbs. O for the sulphur
Total	2.415	lbs. O per lb. of coal
Less	.030	lbs. O already in the coal

Net total = 2.385 lbs. O per lb. of coal to be taken from the air. Atmospheric air contains 23 per cent. oxygen by weight, hence it will require $2.385 \div .23 = 10.37$ pounds of air to completely burn the pound of coal if all the oxygen of the air is used. If 87 per cent. of the pound of coal is

combustible, then there are needed $10.37 \div .87 = 11.91$ pounds of air per pound of combustible.

Where combustion is not perfect the theoretical amount of air is not used. Assume as before that the carbon divides half and half, then we have

For C_1, $.39 \times 2.66 = 1.036$
For C_2, $.39 \times 1.33 = .518$
For H, $.04 \times 8.00 = .320$
For S, $.02 \times 1.00 = .020$

Total 1.894 lbs. O
Less .030 lbs. O in coal

Net Total 1.864 lbs. O to be taken from the air. This makes $1.864 \div .23 = 8.1$ pounds of air per pound of coal burned. Compare this value with that for perfect combustion.

(b) The equation usually quoted for the weight of air needed for perfect combustion is

$$W = 11.52\,C + 34.56\,\left(H - \frac{O}{8}\right) + 4.32\,S \qquad (10)$$

which for the assumed coal is $W = 11.52 \times .78 + 34.56 \left(.04 - \frac{.03}{8}\right) + 4.32 \times .02 = 10.32$ pounds. Compare with the value by chemical analysis.

Probable Amount of Air Used for Combustion (C).—There can be no exact value placed upon actual amount of air passing through a furnace. The construction of the furnace, the type of grate used, the depth of the fuel bed, the quality of the fuel and the eccentricities of the fireman all influence the result. From tests that have been conducted upon various types of heating furnaces under varying conditions of service, it seems reasonable to assume that from two to three times as much air goes through the average furnace as would be needed for perfect combustion. In the most up-to-date power plants excess air is reduced to small amounts.

It is not possible in furnace operation to keep the air supply down to the theoretical amount without reducing the economy of the furnace. When the fuel bed is thick and the air supply reduced, the fuel will receive too small an amount of air and carbon will be burned to CO with a loss of 10150 B. t. u. per pound. When the fuel bed is thin and the supply of air excessive, too much air will pass through the fire causing some of the carbon to pass off unburned and carry-

ing away heat unnecessarily by heating the excess air. (Read Technical Paper No. 137, Bureau of Mines, Washington, D. C.) Of the two alternatives it is better to have too much air than not enough, and some of this air should be admitted above the fuel bed. To illustrate the economy of excess air in practice, suppose the pound of coal just considered is burned in a furnace where the entering air is 60° and the stack gases are 600°. With the specific heat of the gases = .24 we find first, for perfect combustion with 10.37 + .9 = 11.27 pounds of stack gases, the pound of coal has available for boiler use (not counting radiation losses) 13353 — [11.27 × .24 × (600 — 60)] = 11892.4 B. t. u. Second, if there is just enough air to burn the carbon to CO, there will be 6.8 pounds of stack gases and 5436 — [6.8 × .24 × (600 — 60)] = 4555 B. t. u. available. Third, with 2.5 times as much air as is theoretically needed and all the carbon burned to CO_2, there will be 26.83 pounds of stack gases per pound of coal and the heat available will be 13353 — [26.83 × .24 × (600 — 60)] = 9875.8 B. t. u. This shows a decided advantage in favor of excess air over a much restricted supply. Flue gas may be analyzed by the Orsat apparatus and such analysis used in determining the quality of the combustion (See Art. 17).

Theoretical Temperature of the Furnace (D).—When perfect combustion occurs, the theoretical total heat is given off. If it were possible to liberate this heat in a vessel perfectly insulated, all the liberated heat would be used in raising the temperature of the gases. The theoretical rise in temperature in such an ideal furnace would be

$$t_r = \frac{\text{theoretical total heat (B. t. u.)}}{\text{pounds of stack gases} \times \text{specific heat}} \qquad (11)$$

Applying to the coal sample above, t_r = 13353 ÷ (11.27 × .24) = 4946° F., and if the air enters at 60°, the temperature of the furnace is 4946 + 60 = 5006° F.

Probable Temperature of the Furnace (E).—Suppose 2.5 times the theoretical air is used in the furnace, then the probable temperature is

$$t = \frac{13353}{26.83 \times .24} + 60 = 2138^\circ \text{ F.}$$

Radiation and other losses will reduce this value somewhat.

Efficiency in Furnace Combustion (F).—There are five losses in fuel combustion: (a) unburned combustible material that drops through the grate with the ash, (b) unburned

hydrocarbon particles that leave the chimney as smoke, (c) carbon burned to CO instead of CO_2 by incomplete combustion, (d) excessive air supply, (e) radiation. These losses are apportioned about as follows:

(a) (Estimated) 1 to 3 per cent. of total heat in coal.
(b) (Estimated) 1 to 5 per cent. of total heat in coal.
(c) May vary anywhere between 10 and 50 per cent.
(d) May vary anywhere between 5 and 15 per cent.
(e) (Estimated) 2 to 5 per cent.

It will be seen by this that a large part of the original heat in the coal is not transferred through the heating surface of the boiler to the water, but is dissipated through the five channels just mentioned.

Intimately associated with the combustion losses is the idea of *furnace and boiler efficiencies*. The most important of these are grate efficiency, furnace efficiency and overall efficiency.

Grate efficiency =

$$\frac{\text{weight (or heat value) of ascending combustible}}{\text{weight (or heat value) of combustible fired}} \qquad (12)$$

If 2 per cent. of the coal drops through the grate, this is $(100 - 2) \div 100 = 98$ per cent.
Furnace efficiency =

$$\frac{\text{heat available for absorption by boiler}}{\text{heat value of combustible fired}} \qquad (13)$$

With perfect combustion of the entire pound of coal and 2.5 times the required amount of air, this is $9875.8 \div 13353 = 74$ per cent. If there is a percentage loss through the grate, the value 9875.8 will be reduced by this amount. With imperfect combustion, illustrated by the case where the carbon divides half and half to CO_2 and CO, this is $[9395 - 9 \times .24 (600 - 60)] \div 13353 = 61$ per cent. If there is a percentage loss through the grate, the value 9395 will be reduced by this amount.

$$\text{Over-all efficiency} = \frac{\text{heat absorbed by water and steam}}{\text{heat value of combustible fired}} \qquad (14)$$

The heat absorbed by the water and steam is the heat value

of the combustible less all the losses. Suppose the losses in the sample are:

through the grate,	$.02 \times 13353 =$	267.06 B. t. u.;
unburned carbon (smoke),	$.03 \times 13353 =$	400.59 B. t. u.;
imperfect combustion,	$.20 \times 13353 =$	2670.60 B. t. u.;
excessive air supply,	$.10 \times 13353 =$	1335.30 B. t. u.;
radiation,	$.02 \times 13353 =$	267.06 B. t. u.;
total losses,		4940.61 B. t. u.;

then the over-all efficiency is $(13353 - 4940.61) \div 13353 =$ 63 per cent. When the word efficiency is mentioned in connection with small power and heating plants, the *over-all efficiency is understood* unless otherwise specified. The efficiency of the average boiler is 60 to 65 per cent., but efficiencies as high as 75 per cent. may be found in continuous service in some of the better plants (For boiler operation, see Arts. 87 and 187).

17. Flue Gas Analysis.—The quality of the fuel combustion in many plants is determined by the Orsat, or similar apparatus, which is used in obtaining an analysis of the flue gases *by volume* as they leave the boiler. Values are found for CO_2, CO and O. The CO_2 varies from 6 to 17 per cent. of the total volume of the flue gases. Between 10 and 13 per cent. is considered good practice. CO is always found in small quantities, say from 0 to .5 per cent. When excess air is less than 25 per cent., CO is probably forming in prohibitive amounts. With good combustion and 100 per cent. excess air (good boiler practice), there should be but a trace of CO. Free oxygen is always found where there is an excess of air. This percentage of O (0 to 15 per cent.) may be used to determine the amount of excess air.

Of the three determinations made by the use of the Orsat apparatus, the CO_2 and O determinations are considered of greatest value. When carbon and oxygen unite to form carbon dioxide gas, it is found that *with the same temperature and pressure the carbon dioxide occupies the same volume as the oxygen entering into the combination.* Assuming perfect combustion (no carbon monoxide) and just enough air to supply the oxygen, the resulting gas volumes will be 21 per cent. CO_2 and 79 per cent. N. A test with the Orsat in this case should show 21 per cent. CO_2, 0 per cent. CO, and 0 per cent. O. Again, assuming perfect combustion and an excess of air (say 100 per cent.), one-half of the oxygen of the air is used for the CO_2 and the Orsat should show 10.5 per cent.

CO_2, 10.5 per cent. O, and 0 per cent. CO. That is to say, the sum of the CO_2 and O percentages will be 21 per cent., the same as the original oxygen volume. Again, assuming imperfect combustion and a certain amount of CO, it is found that *with the same temperature and pressure the carbon monoxide occupies twice the volume of the oxygen entering into the combination* and the resulting stack gases have a larger volume than the entering air by one-half of the percentage of CO present. With high percentages of CO this change in volume would need to be taken into account. In all ordinary cases, however, it is satisfactory to consider the stack gases as 79 per cent. N and the remaining 21 per cent. composed of CO_2, CO and O. 21 per cent. CO_2 shows the highest possible efficiency, i. e., no excess air and perfect combustion. This is never obtained in practice. Any value of CO_2 less than this indicates (1) excess of air, if no CO is present; (2) deficiency of air, if CO is present and no O; (3) improper mixture in the combustion chamber, if both CO and O are present.

Computations to find the relation between weights of flue gas and entering air are sometimes complicated by the necessity of changing from weights to volumes and vice versa. Volume readings of the Orsat are generally used directly in terms of the densities of the gases since, as above stated, equal volumes of the gases at the same temperature and pressure contain the same number of molecules. Use the equations

$$W = \frac{44\ CO_2 + 32\ O_2 + 28\ (CO + N)}{12\ (CO_2 + CO)} \times C_1 \qquad (15)$$

Where W = weight of flue gas in pounds per pound of coal, C_1 = percentage of carbon in the coal, and the other symbols represent percentages of each as shown by the Orsat. N is found by differences i. e., 100 — $(CO_2 + CO + O)$. For information on the use of the Orsat apparatus see *very excellent explanation in "Coal," by Somermeier.*

Application (1).—Coal, having a composition as stated in Art. 16, is being burned in a furnace without loss through the grate. Samples of the flue gas show 12 per cent. CO_2 and 9 per cent. O. What is the weight of flue gases per pound of coal burned? Compare this value with the theoretical amount of air as in Art. 16 (B) and note the excess supplied. From Equation 15

$$W = \frac{44 \times 12 + 32 \times 9 + 28 \times 79}{12\ (12 + 0)} \times .78 = 16.4$$

Excess air $= 16.4 - 10.37 = 6.03$ pounds. Where a grate loss is known to exist, C_1 should be corrected by this amount. Thus for a 2 per cent. loss, $C_1 = .98 \times .78 = .764$.

APPLICATION (2).—Coal as in application (1); 2 per cent. loss through the grate; 8 per cent. CO_2; 12.5 per cent. O; .5 per cent. CO. Find the weight of stack gases and excess air per pound of coal.

$$W = \frac{44 \times 8 + 32 \times 12.5 + 28\ (.5 + 79)}{12\ (8 + .5)} \times .764 = 22.3$$

Excess air $= 22.3 - 10.37 = 11.93$.

CHAPTER II.

AIR COMPOSITION—VENTILATION—HUMIDITY—DRAFT.

18. Composition of Atmospheric Air:—The subject of ventilation in its relation to health should be introduced by a brief consideration of the properties of the air supplied. Air is a very important factor in building economy. In addition to its value as a heating medium it determines in a marked degree the health of the occupants of the building.

The human body may be considered a well equipped and very complex power plant. As the carbon, hydrogen, and oxygen in the fuel and air supply in any mechanical power plant are consumed in the furnace, the resulting heat absorbed in the generating system and finally turned into work through the attached mechanisms; so the human body absorbs heat from the combustion of food and turns it into work. The products of combustion in both cases are largely carbon dioxide and water. The chief requisites of the mechanical plant are good fuel, well regulated draft and efficient stoking. Similarly, the human body needs good food, pure air and healthful exercise. Of the three requirements all are of the utmost importance, but the second has probably the greatest significance, since no person can long keep in health with impure air, even with the best of food and a sufficient amount of exercise.

In its simplest analysis air is made up of two elements, oxygen and nitrogen, in the volume ratio of 20.9 to 79.1 and a density ratio of 23.1 to 76.9, respectively. In a complete analysis of pure air a number of other elements and compounds are found, making a mechanical mixture that is somewhat complex. Most air samples show traces of carbon monoxide, hydrogen sulphide, ozone, argon, compounds of ammonia, and compounds of nitric, nitrous, sulphuric and sulphurous acids. The heating and ventilating engineer, however, is interested chiefly in the amount of oxygen, moisture and carbon dioxide present. Air taken from the open country and not contaminated with the poisonous gases or the dust and refuse from cities has the following com-

position according to Professor Carpenter, Heating and Ventilating Buildings (See also Encyclopedia Britannica, Respiration).

Oxygen	Per cent.	of	volume	20.26
Nitrogen	"	"	"	78.00
Moisture	"	"	"	1.70
Carbon dioxide	"	"	"	.04

These values are fairly constant, except that of the moisture which may vary from 0+ to 4 per cent. of the entire weight of the air.

Experiments have shown that normally pure air in the process of respiration when exhaled from the lungs of the average person has

Oxygen	Per cent.	of	volume	16
Nitrogen	"	"	"	75
Moisture	"	"	"	5
Carbon dioxide	"	"	"	4

Comparing these values with those for pure air, oxygen is reduced one-fifth, nitrogen is reduced one twenty-fifth, vapor is increased three times and carbon dioxide is increased one hundred times. Oxygen has been consumed in its union with the excess carbon and hydrogen in the human body and is given off as carbon dioxide and water vapor. It may be seen from these ratios, that the gradual reduction of the oxygen content and the very rapid increase of CO_2 with its accompanying impurities soon render unfit for use the air in any building occupied by a number of people. To avoid this state of affairs, fresh air should be supplied continuously and at such points as will provide the most uniform circulation.

19. Oxygen and Nitrogen:—Oxygen fills about one-fifth of the volume in atmospheric air and is the element that makes combustion possible. The other four-fifths of the space is filled with nitrogen. In a providential way this nitrogen acts with the oxygen to control the rapidity with which combustion takes place. Nitrogen seems to have little effect upon respiration, except to retard chemical action. If one were to attempt to live in an atmosphere of pure oxygen, the chemical action taking place through the lungs would be so rapid that the human body would not be able to maintain it.

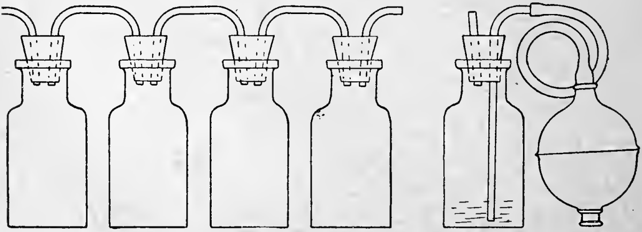
20. Carbon Dioxide:—The amount of CO_2 in the air is used as an index to the purity of the air. It is not consid-

ered a poisonous gas. It has slight taste and odor, but no color. It is found in many natural waters and manufactured beverages, the chief one being "soda water," which is made by forcing carbon dioxide into water under pressure. The real action of CO_2 when taken into the lungs is not well known. It has the effect of producing physical depression and where found in sufficient quantity will even cause death by suffocation, very similar to submergence in water. Whatever its effect upon human life may be, its presence in any room used for habitation (assuming no open fires or gas jets in the room) is an indication of the lack of oxygen and an excess of impurities thrown off by respiration. Good country air has 4 parts CO_2 in 10000 parts of air and room air should never be allowed to have more than 8 to 10 parts in 10000 parts of air. It becomes the duty of the heating engineer therefore to provide pure air in sufficient quantities, to enter and withdraw the air from the room in a manner such as will not be uncomfortable to the occupants and to keep the air fairly uniform in quality throughout the room. Carbon dioxide is 52 per cent. heavier than air of the same temperature and therefore has a tendency to fall. Exhaled air, however, has excessive moisture, has a temperature much higher than that of the room air and is 2 to 3 per cent. lighter than when inhaled. Its tendency to rise neutralizes the excessive density of the CO_2 and as long as the air is absolutely quiet, results in a fair diffusion throughout the room air. In large audience rooms the heat given off from the occupants is sufficient to cause strong currents which carry this impure air to the upper part of the room. A careful study of the physical conditions within inhabited rooms shows that the location of the *bad air zone* may be anywhere from the floor to the ceiling, depending upon the room volume (large or small respectively) allowed per inhabitant and the rapidity of air movement in the room. In investigating air conditions, tests for CO_2 should be made in all sections of the room. Tests conducted at the breathing line represent living conditions for the inhabitants. In residence work air is usually entered and withdrawn at the floor line. In large plants where the air circulates by mechanical means, it usually enters above the heads of the occupants and is withdrawn at the floor line. Some engineers advocate the updraft system with the air entering near the floor and leaving at the ceiling. In the latter case ventilation is simplified but heating is made very expensive.

A method of determining the percentage of carbon dioxide in the air, based upon the fact that barium carbonate is nearly insoluble in water, may be performed as follows: provide eleven bottles with rubber stoppers having two holes each and connect them continuously by glass and rubber tubing, so that if suction be applied at the first bottle of the series air will be drawn in at the last of the series and the same air will be passed through all. In this way a sample of the air to be tested may be drawn into each bottle. The capacities of the bottles in ounces must be respectively, 23½, 18½. 16½, 14, 9½, 7½, 5½, 4, 3¼, 2½ and 2. These may readily be prepared by partially filling with paraffin. Into each bottle is then placed ½ ounce of a 50 per cent. saturated solution of barium hydrate ($Ba(OH)_2$). Air to be tested is drawn through the system until all the bottles contain a fair sample. Each bottle is then thoroughly shaken, so that the liquid may be brought into good contact with the air sample. If the least turbidity or cloudiness appears it indicates

First or largest bottle,	0.04	per cent.	CO_2
Second bottle,	0.06	"	"
Third "	0.07	"	"
Fourth "	0.08	"	"
Fifth "	0.10	"	"
Sixth "	0.15	"	"
Seventh "	0.20	"	"
Eighth "	0.30	"	"
Ninth "	0.40	"	"
Tenth "	0.60	"	"
Eleventh "	0.90	"	"

The glass tubes should extend no farther than the bottom of the stoppers. Fig. 4, *a*, shows four of the bottles and



(a) Fig. 4. (b)

their connections. To illustrate, suppose that the air of a room was tested and that in the first, second, third, fourth, fifth and sixth bottles the liquid became turbid after vigorous shaking. Such room air would have contained 0.15 per cent. carbon dioxide and would have been considered quite unfit for breathing.

A second, less cumbersome method of testing for the percentage of carbon dioxide is shown in Fig. 4, *b*. A bottle of about 6 ounces capacity is fitted with a rubber stopper having two holes. Through one hole a glass tube is brought from the bottom of the bottle and to the outer end of this tube is connected a valved bulb similar to those found on atomizers. Into the bottle are placed 10 cubic centimeters of a solution made by dissolving .53 gram of anhydrous sodium carbonate (Na_2CO_3) in 5 liters of water and adding .01 gram of phenolphthalein. The water used must have been previously boiled for at least one hour in an open vessel. With the apparatus so prepared, squeeze the bulb, thus forcing air from the room through the liquid and into the bottle. The open hole in the rubber stopper is then closed with the thumb and the bottle vigorously shaken. Then another bulb full of air is injected and the bottle again shaken. This process is continued and the number of bulbs of air noted until the red color of the solution, due to the phenolphthalein, disappears. This number of bulb fillings when referred to a table (Similar to Table II) prepared for this particular apparatus, is indicative of the purity of the air. After such an apparatus is completed it must be calibrated before being used. This is done by obtaining the number of bulb fillings of pure country air necessary to clear the liquid, which will usually vary from 40 to 70. The table for use with this special apparatus may be obtained by proportion from Table II, in which the number of bulb fillings of country air is 48. If now with the new apparatus it is found that 60 bulb fillings are required to clear the liquid, the proportionate table would be made by multiplying the number of bulb fillings given below by the ratio of 60 ÷ 48, or 5 to 4. It is important that the bulb be compressed the same amount for each filling, and that the shaking of the bottle and contents be continued the same length of time after each filling, to obtain uniform results. The Wolpert Air Tester, a commercial apparatus, may be obtained for this line of testing.

TABLE II.

Fillings	Per cent. CO_2	Fillings	Per cent. CO_2
48	.030	15	.074
40	.038	14	.077
35	.042	13	.080
30	.048	12	.083
28	.049	11	.087
26	.051	10	.090
24	.054	9	.100
22	.058	8	.115
20	.062	7	.135
19	.064	6	,155
18	.066	5	.180
17	.069	4	.210
16	.071	3	.250

The methods just mentioned for determining CO_2 are fairly satisfactory in obtaining quantitative values from which the quality of the ventilating air in any system may be judged. If exact percentages of CO, CO_2, O and N are required, the Pettersson-Palmquist, the Orsat, or similar apparatus must be employed. For descriptions of these see Stillmans' Engineering Chemistry, Carpenter's Heating and Ventilating Buildings, Hempel's Gas Analysis, translated by Dennis, Abady's Gas Analyst's Manual, and Somermeier's Coal.

21. Amount of Fresh Air Needed per Person:—The need of a continuous supply of fresh air in residences and business houses can scarcely be overestimated. Health is the greatest of all blessings and *pure air is essential to health.* The most convincing argument that can be presented on this point is an analysis of the vital statistics of the country covering a large number of years. Persons afflicted with respiratory diseases are recommended by the medical fraternity to seek a high, dry, sunny climate and *live in the open air.* The rarefied atmosphere causes continuous deep breathing, which exercise in itself has a tendency toward strengthening the afflicted parts and throwing off disease, and the dry air probably serves the lung tissue as a cleanser as the blotter does the page of wet ink. These conditions, in connection with the sunshine which is one of our

best germicides, form the only known remedy for combating such diseases. It is a safe conclusion that the element of pure air which enters so largely into the overcoming of the disease, once it is contracted, is one of the best preventives as well. Statements are made (occasionally in the technical press) that respired air is not harmful and that satisfactory ventilation may be had in inhabited rooms with much less fresh air than that usually allowed. The first of these two statements has never been proved. On the contrary the circumstantial evidence of the impurity of respired air is fairly conclusive. The second may be true for ventilating systems where the air supply is subdivided into small amounts and carried directly to the person (See experiments by Professor Bass at University of Minnesota, Trans. A. S. H. & V. E., Vol. XIX, p. 328). Applications such as this, however, can not be regarded as touching general practice.

The average adult, when engaged in ordinary indoor occupations, will exhale about 20 cubic inches of air per respiration. He will also have 16 to 24 respirations per minute, totaling 400 + cubic inches or, say .25 cubic foot of air per minute. Allowing 4 per cent. CO_2 in respired air the average person will exhale $60 \times .25 \times .04 = .6$ cubic foot CO_2 per hour. This is constantly being diffused throughout the air of the room. If the carbon dioxide and other impurities could be disassociated from the rest of the air and expelled from the room without taking large quantities of otherwise pure air with them, the problems of the heating and ventilating engineer would be simplified, but this cannot be done. Rapid diffusion of respired air throughout the room renders it necessary to dilute the room air with fresh air in order that the purity may be maintained at a safe value. Ideal conditions are found when interior air is as pure and refreshing as that of the open country, but the mechanical difficulties around such a ventilating system would be so great as to render it prohibitive. The standard of purity which should be aimed at, and which may be obtained with a first-class system, is .06 of one per cent. CO_2, i. e., 6 parts of CO_2 in 10000 parts of air. Systems maintaining constant ventilation at 8 parts in 10000 are considered satisfactory. Stated in a simple form for calculation, let Q' = cubic feet of atmospheric air needed per hour per person, A = cubic feet of CO_2 given off per hour per person, n = standard of purity to be maintained (allowable parts of CO_2 in 10000

parts of air), and p = standard of purity in atmospheric air, say 4; then

$$Q' = \frac{A}{n - p} \tag{16}$$

To maintain constant ventilation at 7 parts CO_2 in 10000 parts of air, with pure air at 4 parts in 10000, we have $Q' = .6 \div (.0007 - .0004) = 2000$ cubic feet of air per hour. Based upon .6 cubic foot of CO_2 exhaled per person per hour, Table III gives the amount of air needed to maintain constant ventilation at the various standards of purity.

TABLE III.

Cubic Feet of Air per Person per Hour.

n	A	Q
6	.6	3000
7	.6	2000
8	.6	1500
9	.6	1200
10	.6	1000

It should be understood that no hard and fast rule can be given for the air requirement per person. This varies with the physical development and occupation of the individual, but it varies in a greater degree with the state of the person's health and the sanitary value of his surroundings. In general, the *average adult* subjected to average indoor conditions requires *1800 cubic feet of fresh outdoor air per hour.* Stated as an equation, the amount of air needed for ventilation is $Q' = 1800\ N$, where N = the number of people to be provided for.

The amounts of air in cubic feet per person per hour given in Table IV, may be considered good practice for the various classes of service.

TABLE IV.

Hospitals, Ordinary	2000–2500
" Surgery	2500–3000
" Epidemic	5000–6000
Workshops, Ordinary	1800–2000
" Unhealthy trades	3000–3500
Schools, Offices, Prisons	1800
Theaters and Assembly Halls	1400–1800

One ordinary gas flame of 16 to 20 candle power, using 4 to 5 cubic feet of gas per hour, will vitiate as much air as four or five people. Where many open flame gas lamps are used, this fact should be taken into account.

22. Ventilation:—*Ventilation is the art of maintaining interior atmospheres at a comfortable temperature and humidity, and a purity approaching that of open country air.* Such a standard may be regarded absolutely safe by any one. To accomplish this, large amounts of fresh air should be introduced to the building and distributed so the occupants will not be subjected to unpleasant drafts. Fans placed in the rooms to circulate the air make the room atmosphere more habitable on a warm day, but this process should not be mistaken for ventilation. *The mere process of fanning the air does not purify it.*

Air may be tested for bacteria and micro-organisms by exposing specially prepared gelatine plates or tubes to the air of a room a certain length of time, say five or ten minutes, permitting the organisms to germinate and counting the colonies. (See Report of Ventilation Division, Chicago Health Dept., Page 57, Vol. XX. Trans. A. S. H. & V. E.) Such tests are most satisfactory but require considerable care in application and are not generally used. The CO_2 test mentioned in Art. 20, while not a direct equivalent, is simpler and is generally employed. In testing the quality of room air by any method it is well to call attention to the fact that the ordinary running conditions of any room cannot absolutely be determined by a single test. Trials should frequently be made and records kept. Upon one day atmospheric conditions may be favorable and tests may show a small amount of impurity. On other days when the conditions are not as favorable impurities may be found in large quantities even though running conditions seem to be duplicated. Further, if the only requirement governing the ventilation of buildings is that a satisfactory CO_2 test be passed, there is great danger of overrating or underrating the ventilating system of the building. *A safe method in rating ventilating systems is to require a minimum air supply in addition to a maximum permissible percentage of CO_2 at the breathing line.* For further study of this subject, see recommendations by the American Society of Heating and Ventilating Engineers, Jour. Apr. 1916, p. 91. Also Trans. A. S. H. & V. E.. Vol. XXII, p. 43.

23. Air Purification:—Air contains dust, fine particles of mineral and animal matter, bacteria, and micro-organisms

held in mechanical suspension. The more heavily charged with these impurities ventilating air becomes, the more dangerous it is to the human system. Most materials held in mechanical suspension may be removed by *filtering* (passing through fine cloth screens) or by *washing* (passing through films or sprays of water). Filtering and washing systems are beneficial in all cases and are necessities in many. Filters cost less to install and operate, but they occupy larger transverse areas and are not as effective as the washing systems. Washing air removes most of the mechanically suspended particles but it does not necessarily eliminate chemical impurities, bacteria and the like. The location of the air supply intake to a building carries with it a great responsibility. Air supplied to a building should always be taken from the purest source possible, and when this supply is known to be bad it should be thoroughly washed before sending through the ventilating system.

REFERENCES.—*Trans. A. S. H. & V. E.* Studies in Air Cleanliness, Vol. XXI, p. 211. The Problem of City Dust, Vol. XXI, p. 225.

Ozone is considered by some to be effective as an air purifier. It is an unstable form of oxygen probably containing a greater number of atoms per molecule and is formed by passing air through a highly charged electrical field. Because of its instability as a substance, it readily breaks up and becomes more active as an oxidizing agent than oxygen itself. In its decomposition a part becomes oxygen and the balance is said to enter into combination with substances in the air, thus cleansing the air from these substances. Two claims are made for ozone. The first is that it is a purifier, the second that it is a deodorizer. The first has not been proved satisfactorily, but the second is substantiated by many proofs. Ozone without doubt conceals odors, but it is not known if the substances producing the odors are rendered harmless to the human body.

REFERENCES.—*Trans. A. S. H. & V. E.* An Experiment with Ozone as an Adjunct to Artificial Ventilation at the Mt. Sinai Hospital, N. Y. C., Vol. XXI, p. 256. Air Ozonation, Vol. XX, p. 337. Ozone and Its Applications, Vol. XIX, p. 128. *H. & V. Mag.*, Ozone, July, 1914, p. 16.

24. Moisture with Air:—Moisture in the atmosphere affects the comfort of the occupants as well as the efficiency of the heating and ventilating system in any room. With

moisture in the room a person may feel comfortable when the temperature is several degrees lower than the comfortable temperature of dry air. A dry atmosphere takes up moisture from the room furnishings and from the skin surface of the occupants. The vaporization of moisture from the skin causes a loss of heat from the body and gives to the person a sense of cold which is relieved only when the temperature of the room is increased. An atmosphere that is fairly saturated with moisture demands little evaporation from the skin, in which case the body retains its heat and the person has a sensation of warmth which is relieved only by lowering the temperature of the air of the room. At low temperatures moisture in the atmosphere chills the surface of the skin by actual contact. This is not as noticeable when the air is dry. It follows from the above statements that the range of comfortable temperatures is less for moist

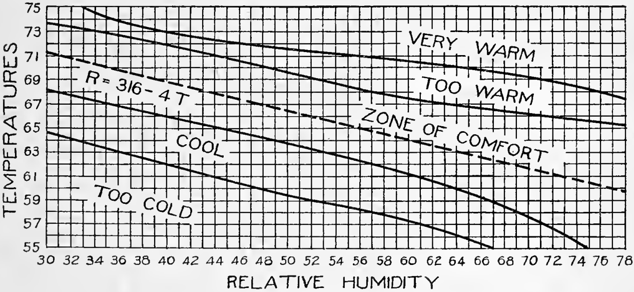


Fig. 5.

air than for dry air. The Chicago Commission on Ventilation, under the direction of Dr. E. Vernon Hill, developed a series of curves from a large number of tests, showing the best relation between the relative humidity and the comfortable temperature in a room (See Trans. A. S. H. & V. E., page 607, Vol. XXIII). The curves in Fig. 5, are plotted from a summary of these tests. It will be noted that the condition represented by 65° and 55 per cent. humidity is as satisfactory as that of 70° and 35 per cent. humidity.

In addition to its effects upon the human body, moisture in the atmosphere has the quality of storing convected heat. It is thus a better heat carrier than dry air and is a benefit to the heating and ventilating system in any building.

REFERENCES.—*H. & V. Mag.* The Primary Physiological Purpose of Ventilation, Sept. 1913, p. 35. *Metal Worker.* Humidity and House Sanitation Explained, Jan. 24, 1913, p. 159. *Trans. A. S. H. & V. E.* The Recirculating of Air in a School Room in Minneapolis, Vol. XXI, p. 109. Relative Humidity, Vol. XVIII, p. 106.

25. Humidity:—*Absolute humidity* is the amount of moisture mixed with the air at any temperature, expressed in grains or in pounds per cubic foot. *Relative humidity* is the ratio of the amount of moisture actually with the air divided by the amount of moisture which the same volume could hold at the same temperature when saturated. The temperature of any air at 100 per cent. saturation (100 per cent. relative humidity) is called the *dew point.* Relative humidity is obtained by using wet-and-dry bulb thermometers or by any one of a number of hygrometers supplied by the trade. The wet-and-dry bulb hygrometer has a very simple application and is generally used. Having given two thermometers (Fig. 6) let one register the temperature of the room air and the other, kept wet by a cloth which covers the bulb and projects into a vessel filled with water, a temperature below that of the room air. If the air is saturated the two thermometers will record the same temperature. If the air is not saturated the thermometer readings will differ according to the humidity. It will be readily seen that the lowering of the mercury in the wet thermometer is due to the extraction of the heat from the mercury column in vaporizing the moisture from the bulb to the air.

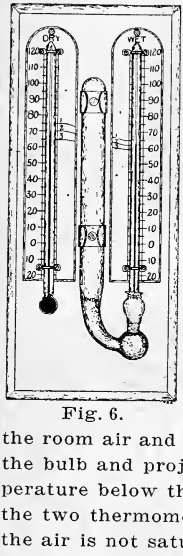


Fig. 6.

In taking readings, let the mercury find a constant level in each thermometer and note the difference in temperature between the two. In Table 12, Appendix, at this difference

and at the room temperature read off the relative humidity. Having found the relative humidity take from Table 13, Appendix, the amount of moisture with saturated air at the temperature recorded by the dry thermometer (absolute humidity at saturation). Multiply this by the relative humidity found and the result is the absolute humidity at the given relative humidity, i. e., the actual amount of moisture with the air per cubic foot of volume.

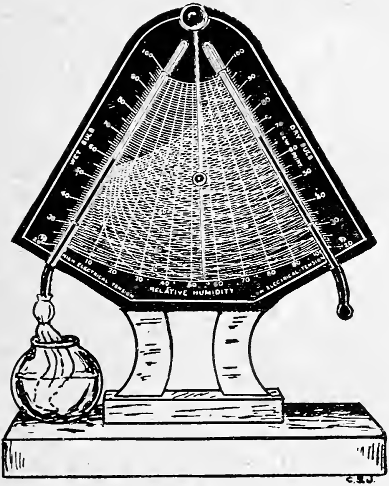


Fig. 7.

Application.—Room air, 70°; difference in readings, 6°. From Table 12, the humidity is 72 per cent. From Table 13, col. 7, .72 × .001153 = .00083 pounds (5.81 grains) per cubic foot.

Instruments have been designed giving the relative humidity by graphical charts. Fig. 7, commonly known as the *hygrodeik*, shows such an instrument. To find the relative humidity swing the index hand to the left of the chart and adjust the sliding pointer to that degree of the wet

bulb thermometer scale at which the mercury stands. Swing the index hand to the right until the sliding pointer intersects the curved line extending downward to the left from the degree of the dry bulb thermometer scale indicated by the top of the mercury column in the dry bulb tube. At that intersection the index hand will point to the relative humidity on the scale at the bottom of the chart. Should the temperature indicated by the wet bulb thermometer be 60° and that of the dry bulb 70°, the index hand will indicate a humidity of 55 per cent. when the pointer rests on the intersection of the 60° wet bulb and 70° dry bulb lines.

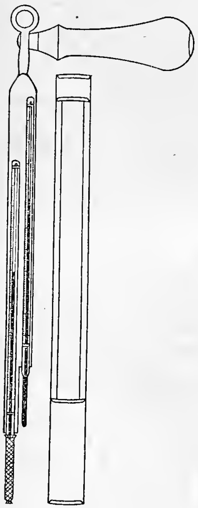


Fig. 8.

The instrument in most general use for humidity determinations is the Sling Psychrometer (See Fig. 8). This is a wet-and-dry bulb outfit pivoted to a handle in such a way that the thermometers may be revolved through the air thus causing a circulation of air over them The wet bulb projects beyond the dry bulb and is covered with a fine mesh cloth. This cloth is dipped into distilled water and the apparatus revolved. Read the mercury level frequently and note the reading of each thermometer at the time the mercury in the wet bulb is at its lowest level. *For accurate work the thermometers should meet a current of air of approximately 15 feet per second,* according to government recommendation.

Table 12, Appendix, represents U. S. Weather Bureau Standards and is used as a reference in this book. Experiments by Mr. Willis H. Carrier, presented in a paper to the American Society of Mechanical Engineers in 1911, show humidities differing somewhat from Table 12 (See "Psychrometric Charts" following Table 14, Appendix).

26. Humidity Chart:—For close approximations the humidity chart (Fig. 9) may be used in determining relative humidity, absolute humidity, dew point, temperature of wet bulb and temperature of dry bulb. On the left of the chart

HYGROMETRIC CHART

GIVING

HYGROMETER TEMPERATURES. RELATIVE HUMIDITY. GRAINS OF MOISTURE PER CU. FT.

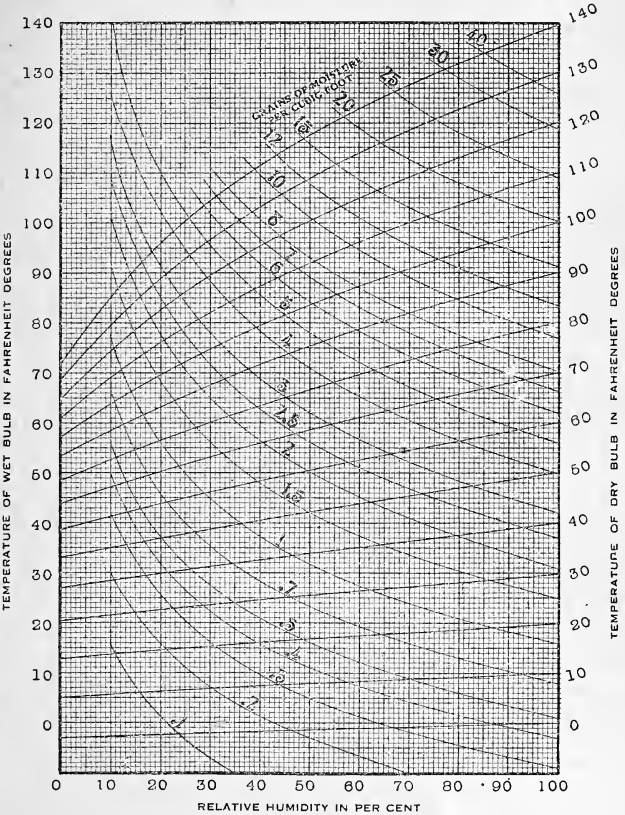


Fig. 9.

Note.—Fig. 9 represents two charts in one. First: the dry bulb temperature curve, which drops to the left, unites with the wet bulb and relative humidity coordinates. Second: the absolute humidity curve, which rises to the left, unites with the dry bulb and relative humidity coordinates. This makes it possible to use the two charts as one, through the relative humidity scale which is common to both.

is a scale referring to horizontal lines giving temperatures of the wet bulb. The scale on the right, referring to the lines curving downward from right to left, is the temperature scale of the room, or dry bulb temperature. The scale along the bottom of the chart gives the relative humidity. The scale of numbers up the center of the chart refers to the lines curving downward from left to right and indicates absolute humidity. For illustration, assume a dry bulb temperature of 70° and a wet bulb temperature 60°, and find relative humidity, absolute humidity and temperature of the dew point. Starting on the right hand scale at 70, follow down the room temperature curve until it crosses the horizontal line of 60° wet bulb temperature. From this intersection drop to the relative humidity scale and read there 55 per cent. To obtain the absolute humidity trace up the relative humidity line to its intersection with the 70° abscissae (horizontal line through 70°) and obtain 4.4 grains per cubic foot. If the room air should drop in temperature, the absolute humidity would remain the same until the dew point is reached (neglecting air contractions). Tracing down the 4.4 grain line to 100 per cent. relative humidity gives the room temperature 52°. This shows that if so cooled the air begins depositing moisture at this temperature. If the temperature of the room air should increase to 90°, the relative humidity may be obtained by following the 4.4 grain line to its intersection with the 90° abscissae line of room temperature and from this intersection dropping to the relative humidity scale at 31 per cent. Thus, having air under any set of temperature and humidity conditions, the effect that a change in any one condition would have upon the others may be obtained without calculations.

Application 1.—The air of a room gives a dry bulb reading of 80° and a wet bulb reading of 69°. What is the relative humidity?

Solution.—Find intersection of dry bulb *curve* and wet bulb *abscissae*. From such intersection drop perpendicular to relative humidity scale and read 57.5 per cent. Check by Table 12, Appendix: 80° room temperature and 11 degrees difference gives 57 per cent. relative humidity.

Application 2.—In the above problem determine the number of pounds of water vapor in the room if its capacity is 3500 cubic feet?

Solution.—At the intersection of the 80° and 58 per cent. *coordinates*, read absolute humidity in grains of moisture per cubic foot as 6.2. Total moisture in room = 3500 × 6.2 = 21700 grains, or 21700 ÷ 7000 = 3.1 pounds of water in form of vapor. Check by Table 13, Appendix. From this table, column 7, the weight of the vapor in pounds present at saturation at 80° is by interpolation, .001578 per cu. ft. At 57 per cent relative humidity each cubic foot would contain .001578 × .57 = .000899 pound and 3500 cubic feet would contain 3.15 pounds.

Application 3.—To what temperature could this room be cooled before moisture would be deposited from the air, i. e., at what temperature of the air would the dew point be reached?

Solution.—The dew point for this room air is the temperature at which 6.2 grains of moisture per cubic foot represents *saturation*, or 100 per cent. relative humidity. Therefore follow the 6.2 grain line to intersection with the 100 per cent. vertical and read 63°. Check by Table 11, Appendix. Temperature at which 6.2 grains moisture becomes the saturation quantity is by interpolation, 62.3°.

Application 4.—To what temperature could this room be heated without moisture addition or loss and maintain a relative humidity of not less than 50 per cent?

Solution.—Following the 6.2 grain line to intersection with 50 per cent. *ordinate*, read from the right the room temperature, 85°. Check by Table 11, Appendix. Since 6.2 grains at the temperature sought will be 50 per cent. of the moisture of saturation at that temperature, 12.4 grains would be saturation quantity, which from Table 11 by interpolation corresponds to 84.2°.

27. Theoretical Amount of Moisture to be Added to Air to Maintain a Certain Humidity:—Warm air has a much greater capacity for holding moisture than cold air. When air of a given outside temperature is heated for interior service, the volume increases with the absolute temperature (See Art. 15). On the other hand, the relative humidity decreases rapidly as shown by the humidity curves (Fig. 9). Air that is dry is unpleasant to the occupants, as well as being detrimental to the furnishings of the room. Therefore, some means should be provided to supply moisture to the incoming air current. In calculating the amount to be

added, let Q = cubic feet of air per hour entering the room at the register temperature t, Q' = corresponding volume at room temperature t' and humidity u', Q_o = corresponding volume at outside temperature t_o and humidity u_o. Also let T, T' and T_o be the absolute temperatures of the entering air, room air and outside air respectively. From the equations

$$TQ' = T'Q \text{ and } TQ_o = T_oQ \tag{17}$$

find Q' and Q_o. From Tables 11 or 13, Appendix, find the amounts of moisture M' and M_o in one cubic foot of saturated air at the temperatures t' and t_o, multiply these by the respective humidities and volumes, and the difference between the two final quantities will be the amount of moisture required per hour as expressed by the equation

$$W = Q'M'u' - Q_oM_ou_o \tag{18}$$

APPLICATION.—Let $Q = 5000$, $t = 130$, $t' = 70$, $t_o = 30$, $u' = .50$, $u_o = .50$, $M' = 7.98$ and $M_o = 1.935$, then

$$Q' = 5000 \times 530 \div 590 = 4490$$
$$Q_o = 5000 \times 490 \div 590 = 4154$$
$$W = 13896 \text{ grains, or } 1.983 \text{ lbs. per hr.}$$

This means that approximately 2 pounds of water would be evaporated for every 5000 cubic feet of fresh air entering the room under the above conditions (See also application in Art. 72).

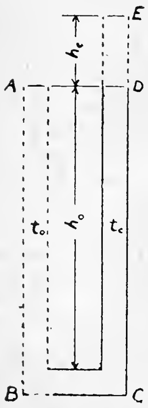


Fig. 10.

28. Velocity in the Convection of Air by the Application of Heat:—Let h_o (Fig. 10) be the height of the chimney or stack. If the temperature of the gases within the chimney CD be the same as that of the entering air there will be no natural circulation, because the column CD will just balance a corresponding column AB upon the outside. If the temperatures of the chimney gases CD and entering air be t_c and t_o respectively, the chimney gases being $(t_c - t_o)$ degrees above that of the outside air, then upon entering the chimney the air becomes less dense and expands according to the ratio of the absolute temperatures before and after heating. With an outside column of h_o feet, it will require a column of the chimney gases $h_o + h_c$ feet to produce equilibrium. In other words, the equivalent column of gases

producing circulation in the chimney has a height of h_c feet. Assume, in the system *ABCDE*, that the interior cross sections at all points are uniform. The volumes of *AB* (imaginary column) and *CE* (actual column) are to each other as their respective heights, and

$V_o : V_o + V_c :: h_o : h_o + h_c$, or $h_o : 460 + t_o :: h_o + h_c : 460 + t_c$. From this we obtain $h_c\ (460 + t_o) = h_o\ (t_c - t_o)$ and

$$h_c = \frac{h_o\ (t_c - t_o)}{460 + t_o} \qquad (19)$$

Substituting for h in the equation $v = \sqrt{2\ gh}$, its corresponding value h_c, we have

$$v = \sqrt{2\ gh_c} = 8.02 \sqrt{\frac{h_o\ (t_c - t_o)}{460 + t_o}} \qquad (20)$$

It is found in practice that the theoretical velocity as given by this equation is never obtained because of the loss of draft due to the friction between the column of gases and the sides of the chimney, and from wind pressures and other causes. Some engineers estimate the actual discharge from the chimney at 50 per cent. of the theoretical. This estimate may be fairly safe for medium sized chimneys but will not be realized on the smaller ones used in residences, which will probably be 25 to 50 per cent. of the theoretical. As the transverse net area becomes smaller, the percentage of friction to the total air moved increases very rapidly and soon becomes the principal factor. Prof. Kent assumed a layer of gases two to three inches thick next to the interior surface as having no velocity and consequently ineffective. Thus a minimum of 4 inches would be added to each theoretical cross dimension to obtain the nominal size of a rectangular chimney.

Some uncertainty will be experienced in the selection of the best values for the *average temperatures* of the chimney gases, t_c, and the outside temperature, t_o, for calculations. t_c is low for residence chimneys because of the low rate of combustion (3 to 7 lbs. per sq. ft. of grate per hr.) and high for large apartment houses, office buildings and power plants (10 to 24 lbs. per sq. ft. of grate per hr.). It is low for unprotected chimneys having large heat loss from radiation and high for those that are housed-in with the build-

ing. Assume $t_o = 70$ for all calculations. Approximate values for chimney height above the grate, h_o, average temperature of gases in chimney, t_c, and temperature of gases entering chimney, t_b, may be taken as in Table V.

TABLE V.

	Residences		Apartment houses	
h_o	30	40	50	60
t_c	200	225	260	300
t_b	300	350	400	450

To estimate the approximate volume of gases circulating through the chimney per second, multiply the pounds of coal burned per hour by 25 (pounds of gases per pound of coal, maximum) times the specific volume of the gas at the temperature of the *entering chimney gases* and divide the result by 3600. Note that the average temperature of the gases is used in obtaining draft but that the entering temperature is used in obtaining area, since all transverse areas are equal and calculated to carry the gases at the entering volume.

When Equation 20 *is applied to hot air stacks in heating systems,* allowances for friction are much less because of the smooth interior of the duct. In such cases the actual velocity of the air should approach more nearly the theoretical. (For applications to chimneys see Arts. 31 and 32).

29. Measurement of Air Velocities:—(See also Arts. 144-146). In ventilating work it is often of the greatest importance to determine air velocities accurately. The correct selection of the sizes of air propelling fans or blowers to do a given work depends largely upon the measurement of the velocity of air delivery. In acceptance and other tests this measurement is equally important.

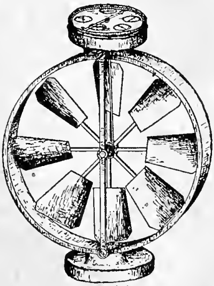


Fig. 11.

Velocities are most commonly measured by means of a vane wheel instrument called the *anemometer*. It is essentially a delicately pivoted wheel having from six to fifteen vanes and similar to the common wind mill wheel (See Fig. 11). To the shaft is connected a recording mechanism consisting of a set of dials which show the velocity of the

air traveling past the instrument. By reading this recording mechanism against a stop watch the velocity of the air per unit of time may be obtained. Since the instrument works against the friction of moving parts its readings are subject to variation and even with frequent calibrations it is not wholly to be relied upon. Various tests of anemometers in comparison with the absolute readings of a gas tank have shown errors as high as 35 per cent. slow to 14 per cent. fast, in the discharge from pipes 8 inches to 24 inches in diameter. It is not fair to condemn a type of instrument because some instruments of the class have failed through long service or lack of care, but in general it is safe to say that the anemometer as an instrument for delicate velocity measurement should be used with great care and should be frequently calibrated.

Velocities are also measured by the *Pitot tube,* Fig. 12. This method of measurement is not as simple as the anemometer but when properly applied it is more accurate. The Pitot tube is essentially a pressure measurer. In every moving fluid (liquid or gas) three pressures are acting. These are commonly designated *dynamic, static and velocity.* Let the

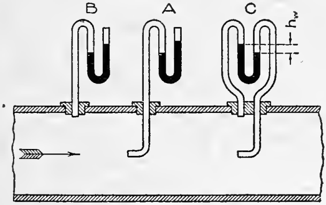


Fig. 12.

bent tube *A* be partially filled with mercury, oil or water as shown and let it be inserted in the pipe with the open end square against the stream. Also, let tube *B* be similarly constructed but let the plane of the opening be 90 degrees to *A*. Tube *A* is acted upon inside the pipe by the atmosphere plus the total forward pressure of the stream (dynamic pressure) and on the outside by the atmosphere. Tube *B* is acted upon inside the pipe by the atmosphere plus the cross pressure (static pressure) and on the outside by the atmosphere. In each case the liquid in the bent tube shows unequal levels, *A* having greater depression than *B*.

Now if the two tubes are united as in *C* so that the pipe pressures act on opposite sides of the same liquid column, the atmospheric pressure is eliminated and the two internal pressures subtract, giving velocity pressure, i. e.,

dynamic pressure — static pressure = velocity pressure.

C shows the instrument as commonly applied. In this the subtraction is automatic and the difference in levels, h_w, is caused by the velocity pressure only. To find the actual velocity of the air in the pipe apply the equation $v = \sqrt{2\ gh}$ where v = velocity in feet per second, g = acceleration of gravity in feet per second, per second and h = the velocity head of the air in feet. If the tube contains water at 60°, the ratio between the specific gravities of air and water being $\frac{62.37}{.0764} = 816.4$ (See Tables 9 and 13, Appendix), the equation reduces to

$$v = \sqrt{2 \times 32.16 \times 816.4 \times h_w \div 12} \qquad \text{or}$$

$$v = 66.2\sqrt{h_w} \tag{21}$$

where h_w = the difference in height *in inches* of the water columns with both legs connected as described and at a temperature of 60°. By a similar method this equation may be deduced for a mercury or other liquid column, or for other temperatures than 60°.

Several Pitot tubes, differing from each other slightly in features of design, are in commercial use. Because of these mechanical differences their readings do not absolutely check each other or those from the theoretical formula, hence all readings must be multiplied by a constant characteristic of the tube in use (See Trans. A. S. H. & V. E., Vol. XXI, p. 459).

In using the Pitot tube or the anemometer, the fact should not be lost sight of that the velocity varies from a minimum at the inner surface of the pipe to a maximum at the center. The friction on the inner surface causes the moving fluid to be retarded next the pipe wall and any test for velocity must account for this variation. With a circular pipe the change of velocity may be approximately repre-

sented by the abscissae of a parabola with its axis on the axis of the circular pipe (See Fig. 13).

The point of *average velocity* is variously quoted from one-fourth to one-third the radius from the wall toward the center, the value depending probably upon the character of the inner surface of the tube. For general use three-tenths will give good average values. For conduits of other shapes the position of mean velocity is difficult to determine. The only safe way is to divide the cross section into small areas and take readings in each area to obtain the average. This

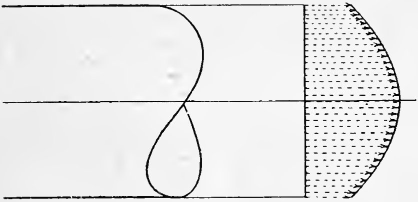


Fig. 13.

variation of velocity from the center of the stream lessening toward the walls may possibly account for many of the variations shown by anemometer tests. It is evident that it is difficult to locate an anemometer so that it will give the correct average reading. In large ducts the error will be less. Pitot tube measurements are more easily applied and are more reliable.

Automatic recording meters may be obtained for keeping permanent records of the flow of air and steam through ducts and pipes. The record from the meter indicates directly the cubic feet of free air or other fluid circulating during each hour of the day.

References.—Kent, *Mechanical Engineers Pocket-Book*. *Trans. A. S. H. & V. E.* Report of the Committee on the Best Way to Take Anemometer Readings, Vol. XIX, p. 202. On Standardization of Use of Pitot Tube, Vol. XX, p. 210. Measurement of Air Flow, Vol. XXI, p. 450. *Trans. A. S. M. E.* Measurement of Air in Fan Work, Vol. XXXIV, p. 1019. The Pitot Tube, Vol. XXV, p. 184. *Jour. A. S. M. E.* Pitot Tubes for Gas Measurement, Sept. 1913, p. 1321.

30. To Determine the Transverse Area of a Chimney for Any Given Height:—The value of any flue as a carrier

of heated gases depends upon both velocity and transverse area. It is not only necessary that a chimney have sufficient height to produce draft but it must have an area capable of carrying the total volume of the gases. The height may be sufficient to create a good velocity but the area may not be sufficient to carry the volume of gases required and the draft becomes ineffective because of clogging. On the other hand, the draft may become ineffective from reduced velocity due to too large an area. In any chimney, height and area are dependent variables. The height is first determined to give a certain draft and to agree with surrounding building conditions, after which the area is determined to carry the gases at the given chimney height and resulting gas velocity. To obtain the *theoretical size* of a chimney, substitute h_o and the assumed values of t_c and t_o in Equation 20 and determine the velocity of the gases per second. Divide the estimated maximum volume of gases moved per second by the velocity to determine the transverse area in square feet and reduce this value to a corresponding round, square or rectangle. For the *actual size* add a minimum of 4 inches to each theoretical dimension.

31. Small Chimneys:—*Application for a ten room residence.* Given: total heat loss from the building per hour 100000 B. t. u., coal 13500 B. t. u. per pound, furnace efficiency 60 per cent. temperature of chimney gases at base of chimney 300°, average temperature of chimney gases 200°, outside temperature 70° and height of chimney 30 feet above the grate.

A heat loss of 100000 B. t. u. per hour will require $100000 \div (13500 \times .60) = 12.35$ pounds of coal per hour at the grate. With gases 300° temperature there will be moved $12.35 \times 25 \times 19.14 = 5933.4$ cubic feet of gases per hour. The velocity of the chimney gases according to equation is 21.8 feet per second, which gives $144 \times 5933.4 \div (3600 \times 21.8) = 10$ square inches, or 3.2-in. × 3.2-in. Adding 4 inches to each dimension = 7.2-in. × 7.2-in., say 8.5-in. × 8.5-in. to fit the brick work. If this were an outside wall chimney it should be 8.5-in. × 13-in.

Application for an apartment house or small school. Given: total heat loss from the building per hour 1000000 B. t. u., $h_o = 60$, $t_c = 300$, $t_b = 450$, $t_o = 70$, and the coal and air conditions as above, find the sizes of the chimney, 8.5-in. × 8.5-

in. (theoretical) and 13-in. × 13-in. (actual). For an outside chimney, at least 13-in. × 17.5-in.

In small chimney construction there is a tendency to leave the interior of the brick work very rough. This should not be, but where such methods are allowed, one dimension of the actual sizes determined as above should be increased by the width of one brick.

32. Large Chimneys:—Chimneys for office buildings, power plants, etc., are generally rated in terms of boiler horse-power. To calculate the sizes of such chimneys, first find the intensity of draft (pressure of the current of gases in inches of water, determined by a draft gage). This will vary from .75 in. to 1.25 in., according to the type of boiler, method of firing, and length and size of breeching. See books on power plant operation. Having the draft, find the height of the chimney, h_o, by the equation

$$d = .52\, h_o\, p_o \left(\frac{1}{T_2} - \frac{1}{T_1}\right) \tag{22}$$

where d = draft in inches of water, p_o = observed atmospheric pressure (commonly taken 14.7), T_2 = absolute temperature of outside air and T_1 = absolute temperature of gases in chimney. Having h_o, find the diameter of a round chimney by the equation

$$\text{B. H. P.} = 2.4\, D^2 \sqrt{h_o} \tag{23}$$

where B. H. P. = nominal boiler horse-power and D = diameter of chimney in feet. For square chimneys find the equivalent area of the round chimney.

Application.—Find the height and diameter of a chimney for 1000 boiler horse-power. Temperature of gases 500°, outside air 70° and required draft, 1-inch of water. In Equation 22

$$1 = .52 \times 14.7\, h_o \left(\frac{1}{530} - \frac{1}{960}\right)$$

$$h_o = 150 \text{ ft.}$$

Also substituting in Equation 23

$$1000 = 2.4\, D^2 \sqrt{150}$$

$$D = 5.8 \text{ ft., say 6 ft.}$$

33. Chimney Notes:—The ideal chimney flue is round in section. Most building construction, however, requires rectangular shapes. These should be kept as nearly square as possible. No chimney flue should be built less than 8-in. ×

8-in. All chimneys should be built up of *hard burned bricks well bedded in cement mortar.* All joints should be struck smooth. *Interiors are improved if lined with hard burned flue tiles.* Chimneys should be built free from other house construction so as to permit the unequal expansion and contraction without cracking the walls of the house or the chimney. The top of the chimney should extend above the highest point of the building. If the top is below any nearby portion of the building, eddy currents will be formed which will enter the top of the flue and seriously reduce the draft. Under such conditions a shifting cowl may be advisable. Chimneys under 30 feet in height are unreliable in their action. Some engineers recommend nothing under 40 feet. The chimney should have no other openings into it than the furnace or boiler smoke pipe. Chimneys in outside walls are not as satisfactory as when built-in, due to the chilling effect of the outside air. When an outside wall chimney is put in it should be made double walled with air space between the walls. A warm air flue by the side of a chimney is an ideal location for the flue. All chimneys should rest upon solid foundations. All joints between the boiler and the chimney should be tight to preserve the draft. Good draft is very essential to the success of any type of heating system, and the purchaser should be required to guarantee a sufficient draft and capacity of his chimney before the manufacturer should be expected to guarantee a stated rating of his furnace, heater or boiler.

References.—Christie, *Chimney Design*, Gebhardt, *Steam Power Plant Engineering*, Marks, *Mechanical Engineers Handbook*, Kent, *Mechanical Engineers Pocket-Book*, *H. & V. Mag.* Baldwin on Chimneys, Oct. 1913, p. 23, Jan. 1914, p. 31.

34. Cowls and Ventilator Heads:—The capacity of any vent or chimney flue may be increased by properly designed cowls surmounting the top of the opening. Much of the down draft experienced under changing wind pressures may thus be eliminated. Shifting heads or cowls take advantage of any wind velocity to increase the upward movement of the air by induction and, when fitted with bearings that permit adjustment from the slightest wind velocity, may be considered highly desirable.

CHAPTER III.

HEAT LOSSES FROM BUILDINGS

35. Heat Dissipated from Buildings:—In planning the heating system for any building, the first and most important part of the work is to estimate the total heat lost in B. t. u. per hour from building. Unfortunately this is the part which is open to the least satisfactory calculation because of varying wind conditions and imperfections in building construction, and because of the lack of accurate conductivity values, especially those relating to the more recent building materials.

Heat is lost from a building in three ways: first, that transferred through the walls, windows and other exposed building materials by conduction and lost by radiation and convection; second, that carried away by convection air currents that pass out through wall cracks and door and window openings to the outside air; third, that lost through specially prepared ventilating ducts. The third item is not included in the usual building heat loss (See Arts. 41 and 42). In the average building the conduction loss is the principal one, although it is now found that the convection loss is of much more importance than has been generally considered. In any case neither of these losses can be determined exactly, but close estimations may be made.

36. Conduction and Radiation Losses:—These losses are considered under various heads, such as glass, wall, floor, ceiling and door losses. Available data have been obtained by experimentation but these do not agree very closely. The reason for so much uncertainty in this part of the heating work is found in the fact that there are great differences in methods of building construction. Conductivity tests on simple materials give fairly uniform results, but when these same materials are assembled in building walls the quality of the workmanship often permits more heat loss by convection than would be transmitted through the materials by conduction. The values quoted for glass and the more compactly built up structures such as brick walls, agree fairly well. The greatest difficulty is found in the balloon frame building with its studded walls, where the dead air space in a well constructed wall may be a good noncon-

ductor, or where on the other hand the same space in a poorly constructed wall may become a circulating air space to cool the walls by the movement of the air.

As an illustration of what may be expected in building losses let Fig. 14 represent a 4-inch studded wall with a tight air space between the studding. It is built up of materials each having a different conductivity and is subjected upon one side to the room temperature t' and upon the other to the outside temperature t_o. Let aa', bb', cc', etc., be planes of equal temperature, but each plane having less intensity of temperature, in the order named, between t' and t_o. Also let the curve xyz represent the temperature drop (measured on the ordinates above an arbitrary zero, not marked) in the heat travel between the entering and leaving radiant heat rays, x and z. It will be noticed that the temperature drop is not uniform along the path of heat travel. This is because of the varying conductivities of the different materials passed through. Along every heat path there are three resistances to the flow of heat between t' and t_o; the air envelope in contact with the wall, the materials composing the wall and the surfaces of each material composing the wall. The summation of these resistances represents the insulating effect against heat flow. It is desirable that these resisting surfaces and materials be of such a character as to cut off heat flow across the wall as completely as possible. The *common defect* found with such wall combinations is *loose construction and air circulation between the studding.* Since the insulating effect of any material or combination of materials is proportional to the total resistance along the heat path, free air circulating between the studding, say from basement to attic, would cause an increased heat loss because the resistances through the latter half of the wall would be eliminated. If, in Fig. 14,

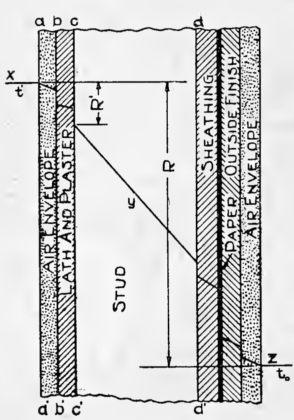


Fig. 14.

the air space marked stud were not tightly closed at bottom and top, the heat crossing from cc' to dd' would be carried away by convection and the insulating qualities of the wall would be R' as compared with R in a tight wall. Still air is a good nonconductor. Convected air is a good heat carrier. Walls of other construction give less uncertainty in heat calculation.

Theoretical equations for heat losses through building walls are based upon conductivity values (reciprocals of resistances per unit thickness) of the various materials and do not take into account such incidental points as intervening air spaces and poor construction. Since the *amount of heat transmitted is equal to the temperature drop divided by the sum of the resistances,* we have for any combination of materials (assuming all surfaces in contact and no air spaces), $H_u = (t' - t_o) \div (R_a + R_b + R_c + \ldots\ldots + R_1 + R_2 + R_3 + \ldots\ldots)$, where R_a, R_b, R_c, etc., are the resistances of the materials and R_1, R_2, R_3, etc., are the surface resistances per unit area. With the material thicknesses m, n, o, etc., and the conductivities K_a, K_b, K_c, ... K_1, K_2, K_3 respectively.

$$H_u = \frac{t' - t_o}{\frac{m}{K_a} + \frac{n}{K_b} + \frac{o}{K_c} + \ldots\ldots + \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}, \text{ etc.}} \qquad (24)$$

Collecting the conductivities in the denominator and placing the reciprocal of this summation as the *combined conductivity* (rate of transmission per unit area), K, we have for any area, A,

$$H = K\ A\ (t' - t_o) \qquad (25)$$

Equation 24 is developed to illustrate a general principle. Its application, however, is usually unsatisfactory and the laborious process is unnecessary when calculating the heat loss for buildings, and Equation 25 is used instead. Values of K commonly used are obtained by experimentation. Table VI has been compiled from a number of the best references.

TABLE VI—Value of K

Materials	K
Brick wall, 8½″ plain	.37
Brick wall, 13″ plain	.29
Brick wall, 17½″ plain	.24
Brick wall, 22″ plain	.21

Brick wall, 27″ plain	.19
Brick wall, furred and plastered, use .7 times non-furred.	
Stone wall, use 1.5 times brick wall.	
Concrete, 2″ solid	.78
Concrete, 3″ solid	.71
Concrete, 4″ solid	.66
Concrete, 6″ solid	.56
Frame wall (plaster, lath, stud, clapboard)	.50
Frame wall (plaster, lath, stud, sheating, clapboard)	.28
Frame wall (plaster, lath, stud, sheating, paper, clapboard)	.23
Windows, single glass, full sash area	1.00
Plate glass, same as single window glass.	
Windows, double glass, full sash area	.50
Skylight, single glass, full sash area	1.10
Skylight, double glass, full sash area	.60
Wooden door, 1″	.40
Wooden door, 2″	.36
Hollow tile, 2″, ½″ plaster, both sides	.41
Hollow tile, 4″, ½″ plaster, both sides	.33
Hollow tile, 6″, ½″ plaster, both sides	.28
Solid plaster partition, 2″	.60
Solid plaster partition, 3″	.50
Concrete floor on brick arch	.20
Fireproof construction as flooring	.10
Fireproof construction as ceiling	.14
Single wood floor on brick arch	.15
Double wood floor, plaster beneath	.15
Wooden beams planked over, as flooring	.17
Wooden beams planked over, as ceiling	.35
Lath and plaster ceiling, no floor above	.62
Lath and plaster ceiling, floor above	.25
Steel ceiling, with floor above	.35
Single ¾″ floor, no plaster beneath	.45
Single ¾″ floor, plaster beneath	.26

Application.—With zero outside temperature the heat losses through the exposed glass and wall surfaces of the Dining Room (Fig. 18), assuming good frame construction, are: glass $= 1 \times 32 \times 70 = 2240$ B. t. u.; wall (minus glass) $= .23 \times 114 \times 70 = 1835$ B. t. u., total 4075 B. t. u. With —10° outside temperature these values are $2560 + 2097 = 4657$ B. t. u.

Most of the values in Table VI have been reduced to chart form (Fig. 15) where the resulting values are the total B. t. u. transmitted through 1 square foot of the surface per hour.

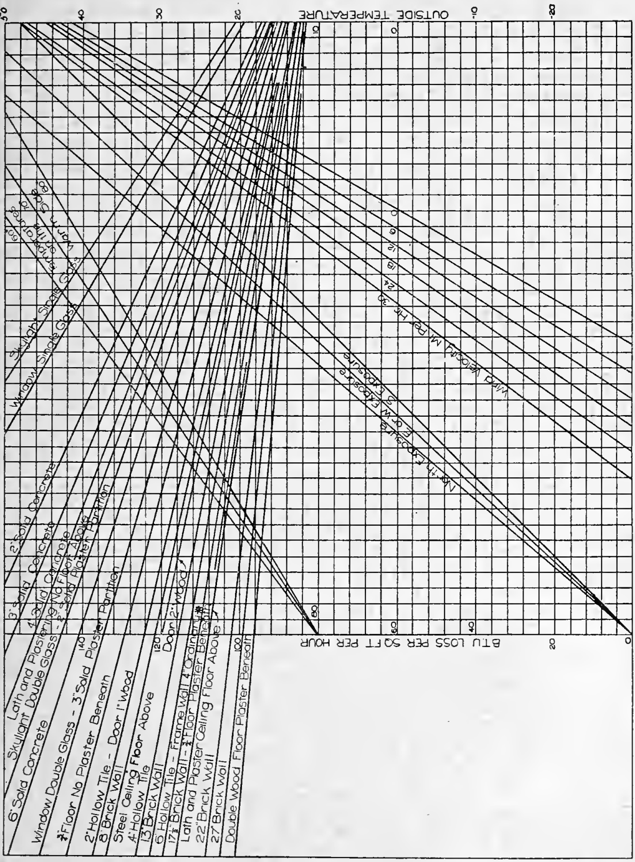


Fig. 15.

APPLICATION 1.—Assume the outside temperature —10°, still air, inside temperature 70° and south exposure. What is the heat loss from a square foot of 13-inch brick wall;

also, from a square foot of single glass window? Beginning at the right of the chart at —10° outside temperature, trace to the left to the 0 wind velocity, then up the ordinate to the 13-inch wall, then to the left to the line indicating 70° inside temperature, then down to the south exposure, then to the left showing 24 B. t. u. transmitted per square foot per hour. For the glass, trace from —10° to the 0 wind velocity, then up to the single window, then to the left to the inside temperature, 70°, then down to south exposure, then to the left showing 80 B. t. u. per square foot per hour. Checking this with the table for a 13-inch brick wall we have, $.29 \times 80 = 23.2$ B. t. u. For glass, $1 \times 80 = 80$. The effect of the wind upon the heat loss is very marked. Locations subjected to high winds should have extra allowances. For example, take the 13-inch brick wall just mentioned. Assume the wind to be 30 miles an hour. By the same process as before we find for a south exposure, 33 B. t. u. loss as compared with 24 at zero wind velocity.

Application 2.—Assume the outside temperature —10°, wind velocity 12 miles per hour, inside temperature 70° and north exposure. What is the heat loss from a square foot of 13-inch brick wall; also, from a square foot of single glass window? Trace as before and find 31 B. t. u. for the wall and 105 B. t. u. for the glass. This is an increase of approximately 30 per cent. over Application 1, due to exposure and wind velocity.

Application 3.—Assume the attic temperature 20°, zero wind velocity, south exposure, room temperature 70°, lath and plaster ceiling with no floor above. What is the heat loss through a square foot of ceiling per hour? Trace from 20° outside temperature and find 32 B. t. u. Checking this with the table, $.62 \times 50 = 31$ B. t. u.

Application 4.—Work out Application 3 for a steel ceiling with floor above and check with the table value.

Application 5.—Assume a 4-inch concrete floor laid on the ground, with a ground temperature of 40° and an air temperature at the floor line of 65°. What is the heat loss through a square foot of the floor per hour? Trace from 40° outside temperature to zero wind velocity, down to 4-inch solid concrete, to the left to 65° temperature, down

to south exposure and to the left to 17 B. t. u. Check by Table VI.

APPLICATION 6.—Assume a 6-inch concrete floor on ground with a ground temperature of 50° and an air temperature at the floor line of 65°. What is the loss through a square foot of the floor per hour? Trace from 50° outside temperature to zero wind velocity (extended), down to 6-inch solid concrete (extended), to the left to 65° temperature, down to south exposure and to the left to 9 B. t. u. Check by Table VI.

37. Loss of Heat by Air Leakage:—Buildings are subject to air leakage through walls, floors, ceilings and window and door clearances. No effort is made to estimate the leakage through walls. In the best type of windows, metal weather strips or other insulations are used. Most of the estimates of building heat losses, however, have to do with ordinary window construction, the quality of the workmanship of which is frequently very poor. Experiments made by H. W. Whitten, R. C. March, S. F. Voohees and H. C. Meyer (*Trans. A. S. H. & V. E.*, Vols. 15 and 22; also, *Jour. A. S. H. & V. E.*, Jan. 1916) to determine the amount of leakage around windows and doors, were very successful in the specific cases. The application of the conclusions to general rules, however, is open to much guess work, since a well fitted window has approximately $\frac{1}{32}$-inch clearance, while a loosely fitted window may have as much as $\frac{3}{32}$-inch. In the tests it was shown also that in any given window clearance the leakage varied greatly as the outside air velocity varied. For illustration, with a clearance of $\frac{1}{16}$-inch the leakage increased 25 per cent. per mile increase of wind velocity; or for a four mile increase in wind velocity the leakage loss increased 100 per cent. With such variations as this the heat loss allowance for the average window leakage is a question. Regardless of the uncertainty in this part of the work, it is interesting to make the best approximation possible and use this in estimating the heat loss from the building.

Some of the approximate values determined by the tests were:

(1) Average wind velocity in localities where heating is important, miles per hr.................... 13

(2) Average sash clearance, in.................................... $\frac{1}{16}$

(3) Air pressure equal to a 15-mile wind against a window having $\frac{1}{16}$-in. clearance will force 146 to 185 cu. ft. of air through each lineal ft. of window clearance per hr. R. P. Bolton recommends 90 cu. ft. Harding and Willard use 60 cu. ft.

(4) Metal weather strips, etc., reduce the leakage as low as 1-5 to 1-9 of that found in the average wood frame window.

(5) The lineal perimeter of the average window is numerically approximately equal to the window area in sq. ft., = G.

From these an estimate may be made for cubic leakage losses through the average window per hour.

Application 1.—What is the window leakage loss from the Living Room, Fig. 18? With a window perimeter = G, a 15 mile wind and a $\frac{1}{16}$-in. clearance we have (assuming 100 cu. ft. per hr. per lineal ft. of perimeter), $42 \times 100 =$ 4200 cu. ft. of air per hr. Since the room is 13′ x 15′ x 10′ = 1950 cu. ft. this leakage would amount to $4200 \div 1950 = 2.15$ room volumes per hr.

Application 2.—What is the leakage loss from

(a)	Dining room?	= 3200 cu. ft. hr. = 1.52	room volumes
(b)	Study?	= 4800 cu. ft. hr. = 2.53	room volumes
(c)	Kitchen? G only	= 3200 cu. ft. hr. = 2.32	room volumes
(d)	Kitchen? G + door	= 5000 cu. ft. hr. = 3.62	room volumes

Professor Carpenter in his heat loss equation makes allowance for leakage losses by using the factors n C for leakage air, in the term $.02\ n\ C$, where n = number of room volumes and C = volume of the room in cubic feet. The use of the term $.02\ n\ C$ is very common practice among heating engineers. The constant .02 is determined as follows: The specific heat of air at 32° is .238; then the number of pounds of air heated from 32° to 33° by 1 B. t. u. is $1 \div .238 =$ 4.2. If the weight of a cubic foot of air at 32° is .0807 pounds, we have $4.2 \div .0807 = 52$ cubic feet of air heated by 1 B. t. u. Since most of the heating is done at an average temperature of 70° the volume of air heated from 69° to 70° by 1 B. t. u. is $52 \times 530 \div 492 = 56$ cubic feet (See absolute temperature, Art. 4). It is evident that some approximation must here be made. No exact value can be taken because

of the great range of temperature change of the air, but 55 is probably the best average. The difficulty of handling the equation with the constant $\frac{1}{55}$ has led to the simple form .02. (See last column Table 13, Appendix).

38. Exposure and Other Allowances:—Air at high velocity passing over the surface of any radiating material is more effective in removing heat than air at low velocity. The north, northwest and northeast in most sections of the country are subject to the highest winds and have the least benefit from the sun, and are therefore counted the cold portions of the building. In estimating heat loss a good way is to figure each room as if it were a south room (assumed to need no additions for exposure) and add a certain percentage of this loss for exposure to fit the real location of the room. The exact amount to add in each case is largely a matter of the judgment of the designer, who of course is supposed to know the direction of the heavy winds and the protection that is afforded by surrounding buildings. Values covering American practice vary between the limits given in Table VII.

TABLE VII.—Exposure.

North, northeast and northwest rooms heavily exposed	10-25 per cent.
East or west rooms moderately exposed	5-15 per cent.
Rooms heated only periodically	20-40 per cent.
Heat interrupted daily but rooms kept closed..	10 per cent.
Heat interrupted daily but rooms kept open....	25 per cent.
Heat off for long periods	50 per cent.
Rooms 12 to 14½ feet from floor to ceiling	3 per cent.
Rooms 14½ to 18 feet from floor to ceiling	6 per cent.
Rooms 18 feet and above from floor to ceiling	10 per cent.

39. Calculation of the Heat Losses—Rule:—Estimate for all rooms to be heated, the number of square feet respectively of exposed glass surface (full sash area), exposed wall surface (gross wall minus glass), exposed doors, floors above unheated or partially heated spaces, ceilings immediately below attic spaces, and partition walls between heated and unheated spaces. With these values and by the use of Table VI, *multiply each surface area by its respective value of K and by the temperature difference between the two air envelopes on the*

sides. To the sum of these products add the amount .02 times the cubic feet of air change per hour times the temperature difference between the inside and outside air, and this will represent the heat loss for a southern exposure. For other exposures add amount allowed for losses due to location from Table VII.

APPLICATION 1.—Referring to Fig. 18, the Living Room will have a heat loss on a zero day as follows: glass, 1 x 42 x 70 = 2940 B. t. u.; wall, .23 x 263 x 70 = 4234.30 B. t. u.; floor (assuming 40° in this part of basement), 45 x 195 x 30 = 2632.50 B. t. u.; and air change (See Table VIII), .02 x 2 x 1950 x 70 = 5460 B. t. u. Total 15267 B. t. u. Since this is a south room there is no allowance for exposure.

The above rule may be stated in equation form. Let H = B. t. u. heat loss from room per hour. With areas in square feet, let G = exposed glass, W = exposed wall minus glass, D = exposed doors, F = floor or ceiling separating warm room from unheated space, etc. Also let $t_x = (t' - t_o)$ = difference between room temperature and outside temperature; $t_y = (t' - t'')$ = difference between room temperature and temperature of the unheated space; K, K' and K'' = coefficients of heat transmission; $Q = nC$ in Arts. 37 and 40 = cubic feet of air change per hour, and a = percentage allowed for exposure. Then

$$H = (KGt_x + K'Wt_x + K''Ft_y + \text{Etc.} + .02\,Qt_x)\,(1 + a) \quad (26)$$

APPLICATION 2.—With same data as in previous application

$$H = (1 \times 42 \times 70 + .23 \times 263 \times 70 + .45 \times 195 \times 30 + .02 \times 2 \times 1950 \times 70)\,(1 + 0) = 15267 \text{ B. t. u.}$$

Good judgment is necessary in selecting the proper outside temperature, t_o, for any locality (See Art. 63). Room temperatures for heated rooms, t', may be taken from Table IX, and temperatures for unheated rooms and spaces from Table X.

Certain credits tending to reduce H are frequently made to the heat loss calculation by allowing for the heat dissipated from lights, persons, etc., within the room (See Art. 44).

40. Short Rules for Estimating Heat Loss:—The method of estimating heat losses outlined in Art. 39 can be recommended for any heat loss calculations. Engineers of experience, however, occasionally develop modified forms for their own use, based upon the method shown in Art. 39 and suited to average building conditions. These short cut methods

should be used with caution by persons not thoroughly acquainted with their development.

Carpenters' Rule.—According to Prof. R. C. Carpenter the quality of building construction and the corresponding heat losses from these buildings are so varied and uncertain that elaborate methods of figuring heat losses are unnecessary. He recommends $K = .25$ for any ordinary wall surface and $G = 1$ for any glass surface. Ceiling and floor surfaces, where it is thought necessary to consider them, may be reduced to equivalent wall surfaces. The rule therefore becomes a simple modification of Equation 26, where $t_x = t' - t_o$.

$$H = (G + .25\,W + .02\,nC)\,(t' - t_o) + \text{exposure} \qquad (27)$$

TABLE VIII—Values of n

Residence heating: halls and bath rooms, 3; living rooms and rooms on the first floor, 2; sleeping rooms and rooms on second floor, 1.	
Offices and stores: first floor, 2 to 3; second floor, 1½ to 2.	The author would suggest that frame construction, large window areas and relatively small volumes tend toward the larger values of n; conversely, brick construction, small window areas and relatively large volumes tend toward the smaller values of n.
Churches and public assembly rooms, ¾ to 2.	
Large rooms with small exposure, ½ to 1.	

With Equation 27, Table VIII should be used and the following wall equivalents may be employed with good effect:

Doors not protected by storm doors or vestibule, with or without small amount of glass = 200 per cent. of equal wall area.

Floors over unheated closed spaces = same as wall.

Floors over partially heated closed spaces = 50 per cent. of equal wall area.

Ceilings below unheated closed spaces, no floors above = 200 per cent. of equal wall area.

Ceilings below unheated closed spaces, floors above = 50 per cent. of equal wall area.

APPLICATION 3.—With the same room and data as in Application 1.

$H = [42 + .25 \times (263 + .5 \times 195) + .02 \times 2 \times 1950]\ 70 =$ 14707 B. t. u.

HARDING AND WILLARD'S RULE.—This is a modification of Carpenter's Rule with the term .02 nC replaced by a leakage factor in terms of the window and door perimeter, P. Use window and door perimeter on that outside wall having the greatest amount of window and door surface.

$$H = (G + .25\,W + CP)\ (t' - t_o) + \text{exposure} \qquad (28)$$

Where the value of C is taken for

Good construction—$\frac{1}{32}$-in. sash clearance....	1.2
Poor construction—$\frac{1}{16}$-in. sash clearance......	2.4
Weather stripped sash	0.15

APPLICATION 4.—With the same room and data as in Application 3, assuming both windows to be affected simultaneously by the air pressure

H, Good Const. $= [42 + .25\ (263 + .5 \times 195) + 1.2 \times 42]$ $70 = 12747$ B. t. u.

H, Poor Const. $= [42 + .25\ (263 + .5 \times 195) + 2.4 \times 42]$ $70 = 16303$ B. t. u.

One of the difficulties in the application of Equation 28 is to determine the character of the sash clearance. In all probability the average value C will approach 2.4 rather than 1.2.

41. Loss of Heat by Ventilation:—Heating and Ventilating systems should have special provisions made for supplying fresh outdoor air for the inhabitants of the rooms and exhausting a corresponding amount of foul air. The exhausted air is usually warm air and as it leaves the rooms carries a certain amount of heat with it. This is a direct loss and should be taken into account.

Since the loss by leakage is the same for any building regardless of the heating system employed, it is accounted for in the ordinary heat loss equation, but losses through ventilating systems must be considered in excess of this amount. Let Q_v = cubic feet of fresh air supplied through the ventilating system per hour, $t' - t_o$ = drop in temperature from the inside to the outside air; then the heat lost by exhausting the air is

$$H_v = \frac{Q_v\ (t' - t_o)}{55} \qquad (29)$$

42. Combined Heat Loss, $H' = (H + H_v)$:—In buildings where ventilation is provided, the total heat loss is that lost by conduction and radiation, H, + that lost by ventilation, H_v (See also Art. 50).

$$H' = H + \frac{Q_v\,(t' - t_o)}{55} \tag{30}$$

Rule.—To find the total heat lost from any building, add to the heat loss calculated by equation, the amount found by multiplying the number of cubic feet of ventilating air exhausted from the building per hour by one-fifty-fifth of the difference between the inside and outside temperatures.

43. Temperatures to be Considered:—In designing heating systems the following temperatures may be used:

TABLE IX—Values of t'.

Living rooms, school rooms, offices, auditoriums, lecture halls and general laboratories	70
Play rooms, gymnasiums, manual training rooms, locker rooms and toilet rooms	65
Bath rooms	80
Hospitals, sick rooms and treatment rooms	75
Greenhouses	70-80
Shops and manufacturing plants, hard labor	60
Shops and manufacturing plants, light labor	65
Paint and finishing rooms	80

Outside temperatures, t_o, should be estimated from the lowest temperature recorded by the weather bureau for that locality, during the preceding ten years. This will range from 10° in the southern to —30° for the northern sections of the country. The most extreme low temperatures are of such short duration that one is not justified in designing for these. Usually ten degrees above the lowest recorded temperature is used (See Art. 63).

The temperatures of rooms not specifically heated may be taken:

TABLE X—Values of t_o when applied to a room

Cellars and rooms kept closed	32
Rooms often in communication with the outside air, such as passages, entrance halls, vestibules, etc	23

Attic rooms immediately beneath metal or slate roof	14
Attic rooms immediately beneath tile, cement, or tar and gravel roof	23

44. Heat Given Off from Lights and from Persons Within the Room:—As a credit to the heating system, some heating engineers take account of the heat radiated from lights and persons within the rooms. The following values are collected from various authorities and may be considered fair averages:

TABLE XI.

Gas, ordinary split burner, B. t. u. per candle power hr.	300
Gas, Argand " " " " "	200
Gas, Auer " " " " "	31
Petroleum " " " " "	160
Alcohol, incandescent " " " " "	40
Electric, incandes'nt carbon filament " " "	14
Electric, " metal filament " " "	4
Electric, arc	5

According to Pettenkofer, the mean amount of heat given off per person per hour is 400 heat units for adults and 200 for children.

45. Performance to Guarantee Heating Capacity:—Some contracts guarantee that the heating system (steam or hot water radiation) will maintain the interior temperature of the building at 70° when the outside temperature is zero or some value below. It is frequently necessary to make tests to prove the fulfillment of such guarantees when the outside temperature is above that stated in the guarantee. It is evident that the inside temperature of the room while under test will then be in excess of 70°. To maintain the temperature that will give an equivalent heating value to the guarantee, is the object of the test. Tests of this character are never as satisfactory as when conducted under guaranteed conditions, but may be estimated with a fair degree of accuracy. *A method proposed by William Kent* in the Engineering Record, Aug. 11, 1894 (See also M. E. Pocket Book), assumes that K is constant for any given material under temperature differences ordinarily found in practice; also, that the heat lost from the house equals the heat given

up by the radiator. It is found from experimental data that K is not constant for varying temperature differences but that it may be so considered without serious error.

Let R = sq. ft. of radiator surface; W_b = sq. ft. of surface of exposed walls, windows, etc.; t_s = temperature inside the radiator; t' = room temperature while under test; t = guaranteed room temperature; t'_o = outside temperature at time of test; t_o = outside temperature specified on guarantee; K_r = rate of transmission through radiator; K_b = average rate of transmission through building walls. From the conditions of guarantee $K_r R (t_s - t) = K_b W_b (t - t_o)$; $c = (K_b W_b \div K_r R)$; $t = (t_s + ct_o) \div (1 + c)$ and $c = (t_s - t) \div (t - t_o)$. Then from the conditions of the test

$$t' = (t_s + ct'_o) \div (1 + c) \qquad (31)$$

which gives the temperature of the room under test corresponding to the given values of t_s and t_o.

Application 1.—Suppose the heating system in any design is guaranteed to heat the interior of the house to 70° at —10° outside temperature, when the steam pressure is atmospheric, and that the test of acceptance is to be run when the outside temperature is 60°. What will be the maintained inside temperature, t', to satisfy this guarantee? From the conditions of the guarantee find $c = (212 - 70) \div [70 - (-10)] = 1.775$. Then from the conditions of the test $t' = (212 + 1.775 \times 60) \div (1 + 1.775) = 115°$. In this same application if the heating system is guaranteed to heat to 70° when the outside temperature is 0° we would have $t' = (212 + 2.029 \times 60) \div (1 + 2.029) = 110°$.

A *second method*, very similar to the preceding and found in Mechanical Equipment of Buildings, Vol. 1, Harding and Willard also makes the assumption that K is constant for varying temperatures. From the two equations, $(G + .25\,W + .02\,nC)\,(t - t_o) = K_r R\,(t_s - t)$ and $(G + .25\,W + .02\,nC)\,(t' - t'_o) = K_r R\,(t_s - t')$, we have by division $(t' - t'_o) \div (t - t_o) = (t_s - t') \div (t_s - t)$ and

$$t' = \frac{t_s\,(t'_o + t - t_o) - t'_o \times t}{t_s - t_o} \qquad (32)$$

Application 2.—With the same conditions of guarantee and test as given in Application 1. $t' = 115°$ for $t_o = -10°$ and 110° for $t_o = 0$.

A *third method*, by W. W. Macon, is shown in Table 48, Appendix.

CHAPTER IV.

FURNACE HEATING AND VENTILATING.

PRINCIPLES OF DESIGN.

46. Furnace System Compared with Other Systems:— The plan of heating residences and other small buildings by furnaces in which the air serves as a heat carrier, is common in this country. Some of the *points in favor* of the furnace system are: low cost of installation, heating combined with ventilation, and adaptability to light service and sudden changes of outdoor temperature. Compared with that of other heating systems, a first-class furnace system can be installed for one-third to one-half the cost. In addition to this, the fact that ventilation is so easily obtained and that the consumption of fuel may be so nearly proportioned to the demands of the weather, give this method of heating many advocates. The *objections* to the system are: the difficulty of heating the windward side of the house, circulated dust, and the contamination of the air supply by the fuel gases leaking through the joints in the furnace. In a good system well installed, the only objection to be seriously considered is the difficulty of heating that part of the house subjected to the pressure of the heavy wind. The natural draft from a warm air furnace is not very strong at best and any differential pressure in the various rooms will tend to force the air toward those rooms offering the least resistance. In a properly designed furnace plant, however, the layout may be so made as to reduce this possible differential to a minimum. The *cost of operation* can be largely controlled by the owner, consistent with his ideas of the quality of the ventilation needed. Arrangements may be made to carry the room air back to the furnace to be reheated, in which case (fresh air cut off entirely) the cost of heating is about the same as that of any system of direct radiation having no special provision for ventilation. Beyond this, any amount of fresh air desired may be taken from the outside and mixed with the room air for the purpose of ventilation. This

requires the same amount of room air to be exhausted from the house at the room temperature and causes an increased cost of operation, as discussed in Art. 50.

47. Essentials of the Furnace System:—Fundamentally this installation must contain a furnace upon a proper setting, a carefully designed and constructed system of fresh air supply and return ducts, and the warm air distributing leaders, stacks and registers. Fig. 16 shows a common

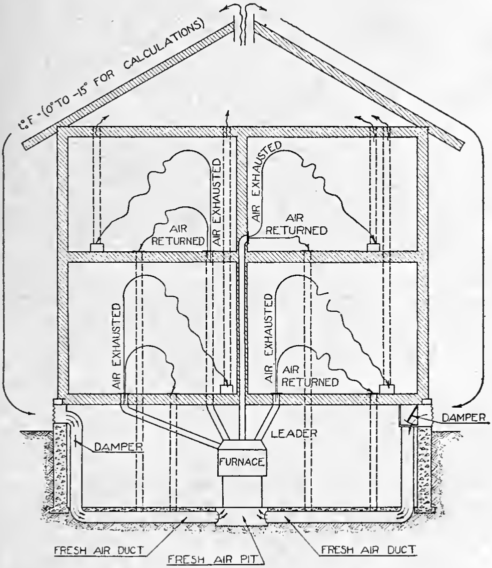


Fig. 16.

arrangement of these essentials. Dampers in the various air lines in the basement provide means whereby fresh air may be taken from the outside or recirculated air from the rooms as desired. Return registers and ducts are placed in the coldest sections of the building (in some cases each room) and should lead by the shortest lines to the furnace.

48. Points to be Calculated in a Furnace Design:—In addition to the calculated heat loss, H, which may be assumed the same for all methods of heating, other points in

furnace plant design should be taken up in the following order: find for each room the cubic feet of air needed as a heat carrier and determine if this amount of air is sufficient for ventilation; then obtain from this the areas of the net heat registers, gross heat registers, heat stacks, net vent registers, gross vent registers, vent stacks, leader pipes, fresh air duct and total grate area. From the total grate area select the furnace.

49. Air Circulation in Furnace Heating:—The use of air in furnace heating may be considered from two standpoints, each very distinct in itself. First, *air as a heat carrier;* second, air as a health preserver. The first may or may not be fresh air. All that is necessary is to provide enough air to carry the required amount of heat from the furnace to the rooms, i. e., that amount of heat that will replace the heat lost by radiation plus the small amount that is carried away by leakage. With given temperatures of air at the register and in the room, the volume of air (volume at the register) may be easily calculated. The second requires that enough air be sent to the rooms to provide ventilation for the occupants. Each of these two amounts should be determined and the greater used in estimating the sizes of the registers and ducts. As previously stated, the cubic feet of air per hour for ventilation may be taken 1800 N, where N is the number of persons to be provided for (See Art. 21).

50. Air Circulated per Hour and Total Heat Loss:—A safe temperature t, of the circulating air as it leaves the heat register, is 130°. This may at times reach 150° or above, but it is not well to use the higher values in the design calculations. If the room air temperature is $t' = 70°$, the incoming air to the room will drop in temperature through 60 degrees, and since one cubic foot of air can be heated through 55 degrees by one B. t. u. it will give off $60 \div 55 = 1.09$ (say 1.1) B. t. u.

Let Q = cubic feet of air per hour as a heat carrier; H = total heat loss in B. t. u. per hour by equation; t = temperature of the air at the register; and t' = temperature of the room air; then

$$Q = \frac{55\ H}{t - t'} \tag{33}$$

Rule.—To find the cubic feet of air necessary to carry the heat to the rooms, multiply the heat loss calculated by equation by fifty-

five and divide by the difference between the register and the room temperatures. For ordinary furnace work this becomes

$$Q = \frac{H}{1.1}$$

Now if this air is not specially exhausted from the building but is taken back to the furnace and recirculated, the only loss of heat will be H. Since air thus used would soon become unfit for the occupants to breathe, it is well to exhaust through ventilating flues a part or all of the air sent from the furnace. This makes an additional loss of heat corresponding to the drop in temperature from 70° to that of the outside air (See Arts. 41 and 42). If the temperature of the outside air is assumed 0°, —10° and —15° respectively, the resulting heat loss will be

$$H'_0 = H + 1.27\ Q_v;\ H'_{-10} = H + 1.45\ Q_v;\ H'_{-15} = H + 1.54\ Q_v \quad (34)$$

For illustration, consider the Living Room (Fig. 18) under three conditions on a zero day: first, when all the air is recirculated; second, when only enough air is exhausted to give fresh air for ventilation; third, when all the air is exhausted. Under the first case the loss H is 15267 B. t. u. per hour and no other loss is experienced. In the second case, if three people occupy the room and each is allowed 1800 cubic feet of fresh air per hour, the total heat loss will be $15267 + 1.27 \times 5400 = 22125$ B. t. u. In the third case, where all the air is exhausted, $15267 \div 1.1 = 13879$ cubic feet of fresh air per hour will be raised from 0° to 70° which will increase the heat loss $1.27 \times 13879 = 17626$ B. t. u., making a total loss of 32893 B. t. u. per hour. The second condition is that which would be found most satisfactory.

It is evident from inspection that the cubic feet of air necessary as a heat carrier will be excessive for ventilation in the average residence (See Art. 51), and the designer need not consider the amount of air for ventilation in calculating the sizes of the ducts and registers. However, this will be needed in an investigation of the size of the furnace, the amount of coal burned or the cost of heating; the latter being in direct proportion to the respective total heat losses (See Art. 63).

Application.—Referring to Table XII, the calculated amount of air Q, for the various rooms of a residence may be found.

51. Is This Amount of Air Q, Sufficient for Ventilation if Taken from the Outside?—Assume the same room as in Art. 50 with $Q = 13879$ cubic feet. With a room volume of 1950 cubic feet, the air will change 7.1 times per hour and, allowing 1800 cubic feet of air per person, will supply eight persons with good ventilation if fresh air is used. Stated as an equation this is

$$N = \frac{H}{1.1 \times 1800} = \frac{H}{2000} \text{ approx.} \tag{35}$$

As a matter of fact, ventilation for half this number will be ample in an ordinary residence room excepting on extraordinary occasions. Test Q for other rooms and find that *ducts and registers designed sufficiently large to carry air for heating purposes are ample for ventilation in residences.*

52. Given the Heat Loss H and the Volume of Air Q' for Any Room, to Find t, the Temperature of the Air Entering at the Register:—If for any reason Q is not sufficient for ventilation (schools, offices, auditoriums, etc.), more air must be sent to the room and the temperature dropped correspondingly to avoid overheating the room. Let Q' = total volume of air per hour (including extra air for ventilation), measured at the register, then

$$t = 70 + \frac{55\,H}{Q'} \tag{36}$$

Rule.—When it is necessary for ventilating purposes to circulate more air than that calculated from the heat loss equation, then the temperature at the register will be found by adding to seventy degrees the amount found by multiplying the heat loss by fifty-five and dividing by the cubic feet of circulated air. This rule applies to all indirect heating.

Application.—Suppose it were necessary on a zero day to send 18000 cubic feet of fresh air to the above room per hour to accommodate ten people. The temperature of the air at the register should be

$$t = 70 + \frac{55 \times 15267}{18000} = 116.6°$$

53. Net Heat Registers:—The velocity of the air, V, as it leaves the heat register is assumed 3 to 4 feet per second by different designers. The mean value is recommended for registers placed near the floor line. Where they are placed

above the heads of the occupants of the room (See Art. 134), higher velocities may be used. The general equation for net register area in square inches is

$$N.\ H.\ R. = \frac{H \times 55 \times 144}{(t - t') \times v \times 3600} \tag{37}$$

Rule.—To find the square inches of net heat register, multiply the heat loss calculated by equation by two and two-tenths and divide by the product of the velocity in feet per second times the difference in temperature between the register and the room air.

Assuming a mean velocity of 3.5 feet per second for all floors and 60 degrees drop in temperature from the register to the room, Equation 37 becomes

$$N.\ H.\ R. = \frac{H \times 55 \times 144}{60 \times 3.5 \times 3600} = .01\ H \tag{38}$$

54. Net Vent and Return Registers:—Vent registers or return registers or both should be put in at every important part of the design, but this is not always done. In order that any room may be heated properly it is necessary that the room air be allowed to escape to permit the heated air to come in. This may be done by venting through doors, windows or transoms but the ideal way is through special ducts to the attic or back to the furnace. A tightly closed room cannot be properly heated by a furnace (See Art. 67).

If all the air were to pass out the vent or return register, at the same velocity as it entered through the heat register, the area of the vent or return register would be to the area of the heat register as the ratio of the absolute temperatures of the leaving and entering air; that is, the area of the vent or return register = .9 the area of the heat register. Since some of the air leaves the room through other openings, these registers need not be so large, say

$$\left.\begin{matrix} N.\ V.\ R. \\ N.\ R.\ R. \end{matrix}\right\} = .007\ H = .7\ N.\ H.\ R. \tag{39}$$

55. Gross Register Area:—The nominal size (catalog size) of the register is usually stated as the two dimensions of the rectangular opening into which it fits. The area of this opening varies from one and one-half to two times the net area. The larger value is for floor registers and is the safer one to follow unless the exact value is known for any

special make of register. Wall registers have lighter bars and for the same net area have somewhat smaller gross area.

$$G.\ R. = (1.5 \text{ to } 2) \text{ times the net register} \qquad (40)$$

Round registers may be had if desired. Register sizes may be found in Tables 19 and 21, Appendix.

56. Heat Stacks:—The vertical ducts delivering air to the registers are called *stacks*. To install the proper sized stacks in any heating system is very important. By some designers the cross sectional area is taken a certain ratio to that of the net register. This has been quoted anywhere from 50 to 90 per cent. Prof. Carpenter suggests 4, 5, and 6 feet per second respectively, as the air velocities for stacks leading to the first, second, and third floors. Mr. J. P. Bird (Metal Worker, Dec. 16, 1905) used 280, 400, and 500 feet per minute, which is approximately 4.5, 6.5, and 8 feet per second for the respective floors. The cross sectional areas of the heat stack, with velocities 4, 5.5, and 7 feet per second, are

$$\text{H. S.} = \frac{H \times 55 \times 144}{60 \times (4, 5.5, \text{or } 7) \times 3600} = \begin{array}{l} .0091\,H \text{ 1st floor} \\ .0066\,H \text{ 2nd floor} \\ .0052\,H \text{ 3rd floor} \end{array} \qquad (41)$$

Rule.—See rule under net heat registers with changed value for velocity.

The theoretical air velocity in the stack is based upon the equation $v = \sqrt{2gh}$, where h = (effective height of stack) $\times$ $(t - t') \div (460 + t')$; v is in feet per second; t is the temperature of the stack air; and t' is the temperature of the room air. The calculated velocities from this equation are much higher than those that obtain in practice because of the retarding influence of the shape of the cross section, the friction of the sides, and the abrupt turns in the stack.

Assuming the net register to be figured at 3.5 feet per second, the quotations by Carpenter and Bird give heat stack areas for the first floor, 88 and 75 per cent.; second floor, 70 and 53 per cent.; and third floor, 58 and 42 per cent. of the net register. Good sized stacks are always advisable (See Art. 71, but because of the limited space between the studding it becomes necessary at times to put in a stack that is too small or to increase the thickness of the wall, a thing which the architect is occasionally unwilling to do. From

the above figures, checked by existing plants that are working satisfactorily, the following approximate figures will give good results.

$$H.\ S. = \begin{cases} .008\,H = .8\,N.\,H.\,R.,\ \text{first floor} \\ .006\,H = .6\,N.\,H.\,R.,\ \text{second floor} \\ .005\,H = .5\,N.\,H.\,R.,\ \text{third floor} \end{cases} \quad (42)$$

57. Vent and Return Stacks:—Estimated in the same manner as the $N.\ V.\ R.$, these may be made

$$\left.\begin{matrix} V.\,S. \\ R.\,S. \end{matrix}\right\} = .7\,H.\,S. \quad (43)$$

As a matter of practice it will be satisfactory to make these stacks in average residence rooms, one or more tin stacks, full opening between studs; the total cross sectional area approximating the equation.

58. Leader Pipes:—Since all the air that passes through the stacks must pass through the leader pipes, it might be assumed that the cross sectional areas of the two would be equal. There are two reasons why this should not be. Because of their vertical position, stacks offer less frictional resistance, area for area, than leader pipes with their small pitch and abrupt turns. Also there is some drop in temperature as the air passes through the leader pipes, consequently the volume entering from the furnace is greater than that going up the stack. Considering these points it would be well to make the area of the leaders

$$L.\ P. = \begin{cases} (.008 \text{ to } .009)\,H = (.8 \text{ to } .9)\,N.\,H.\,R.,\ \text{first floor} \\ (.006 \text{ to } .007)\,H = (.6 \text{ to } .7)\,N.\,H.\,R.,\ \text{second floor} \\ (.005 \text{ to } .006)\,H = (.5 \text{ to } .6)\,N.\,H.\,R.,\ \text{third floor} \end{cases} \quad (44)$$

the exact figures to depend upon the length and inclination of the leader (See Art. 69).

59. Fresh Air Duct:—The area of the fresh air duct is determined largely by experience as in the case of the vent and return lines. It is generally taken

$$F.\ A.\ D. = .8 \text{ times the total area of the leaders} \quad (45)$$

Assume the average velocity of the air in the leaders to be 6 feet per second and the area of the fresh air duct to be as stated, then if the air in each were of the same temperature, the velocity in the fresh air duct would be $6 \div .8 = 7.5$ feet per second; but since the temperatures are different, the velocities will be in proportion to the absolute temperatures. In this case— 0°, $.78 \times 7.5 = 5.8$; at 25°, $.82 \times 7.5 = 6.2$; and at 50°, $.88 \times 7.5 = 6.6$ feet per second. It is seen by this that although the area of the fresh air duct is contracted to

80 per cent. of that of the leaders, the velocity is below that of the leaders. It is always well to have a fresh air duct that is simple in cross sectional area and free from obstructions and sharp turns.

60. Grate Area:—The grate area of a furnace is estimated from the total heat loss, assuming the quality of the coal, the efficiency of the furnace, and the pounds of coal burned per hour per square foot of grate. The heat value of the coal will be between 11000 and 14000 B. t. u. per pound as shown in Table 15, Appendix. The efficiency of the average furnace is approximately 60 per cent., and the coal burned per square foot of grate per hour ranges from 3 to 7 pounds (See Art. 61). Furnaces are charged from two to four times each twenty-four hours. This requires a good sized fire pot and a possibility of banking the fires. To allow 5 pounds per square foot of grate per hour is as good an average as can be made for most coals in furnace work. Let H' = total heat loss from building including ventilation loss, E = efficiency of furnace, f = value of coal in B. t. u. per pound, and p = pounds of coal burned per square foot of grate per hour. The equation for the square inches of grate area is

$$G.\ A. = \frac{H' \times 144}{E \times f \times p} \tag{46}$$

Rule.—To find the square inches of grate area for any furnace, multiply the total heat loss from the building per hour by one hundred and forty-four and divide by the quantity found by multiplying the total pounds of coal burned per hour by the heat value of the coal and the efficiency of the furnace.

Application.—In the typical residence (Art. 62), H on a zero day is 110574 B. t. u. per hour. This will require 101000 cubic feet of air per hour as a heat carrier. Assuming as a maximum that ten people will be in the house and that they will need 18000 cubic feet of fresh air per hour for ventilation, this air will carry away approximately 22900 B. t. u. per hour, making a total heat loss from the building of 133474 B. t. u. per hour. If the furnace is 60 per cent. efficient and burns 5 pounds of 14000 B. t. u. coal per hour per square foot of grate, we have

$$G.\ A. = \frac{133474 \times 144}{.60 \times 14000 \times 5} = 458 \text{ square inches} = 24 \text{ inches}$$

diameter. With coal at 13000 B. t. u. per pound, the grate

would be 493 square inches or 25 inches diameter; at 12000, 534 square inches or 26 inches diameter; at 11000, 582 square inches or 27 inches diameter.

In any specific case it would be wise to estimate the grate size from the heat value of the poorest grade of coal likely to be used. In this case the estimated diameter of the grate varied three inches between coal samples nominally rated at 14000 and 11000. This variation is too great to be overlooked in the selection of furnaces. With the assumption made above, the equation becomes $G.\ A. = .0035\ H'$ for the better grades of coal, and $G.\ A. = .0044\ H'$ for the poorer grades. For the average coals a fairly safe value is

$$G.\ A.\ \text{square inches} = .004\ H' \qquad (47)$$

61. Heating Surface:—The right amount of heating surface to require in any furnace is rather an indefinite quantity. Manufacturers differ upon this point. Some standards may soon be expected but at present only rough approximations can be stated. One of the chief difficulties is in determining what is, or what is not, heating surface. Some quotations no doubt include surfaces that are very inefficient. In estimating, only prime heating surface should be considered, i. e., plates having direct contact with the heated flue gases on one side and the warm air current on the other. If these plates transmit K, B. t. u. per square foot per degree difference of temperature, t_z, per hour; and if one square foot of grate gives to the building $E \times f \times p$ B. t. u. per hour, there will be the following ratio between the heating surface and grate surface:

$$\frac{H.\ S.}{G.\ S.} = \frac{E\, f\, p}{K\, t_z} \qquad (48)$$

APPLICATION.—With $K\, t_z = 2500$ (*Trans. A. S. H. & V. E.*, Vol. XII, p. 133; also, *Jour. A. S. H. & V. E.*, Jan. 1916) and the same notations as in Art. 60.

$$\frac{H.\ S.}{G.\ S.} = \frac{.6 \times 14000 \times 5}{2500} = 17$$

In practice this ratio varies anywhere betwen 12 and 30.

From investigations by the Federal Furnace League (now The National Warm Air Heating and Ventilating Association), furnaces showed an average of 1½ square feet of direct heating surface and 1 square foot of indirect heating surface, making a total of 2½ square feet of average heating surface per pound of coal burned in the furnaces per hour.

In the tests of these furnaces combustion rates as high as eight pounds of coal per square foot of grate were obtained. At this rate of burning the ratio of the heating surface to the grate surface is 20 to 1. It is the opinion of the author that although good service is obtained in tests by combustion rates as high as eight pounds, furnaces should be selected at a lower value, say five pounds.

62. Application of the Above Equations to a Ten Room Residence:—In every design, complete calculations should be made and the results tabulated for easy reference and comparison. Such a tabulation is shown in Table XII, which gives all the calculated quantities (in some cases modified to suit standard sizes) necessary in the installation of the furnace system illustrated in Figs. 17, 18 and 19. The value of condensing the work in this way facilitates checking and the detection of errors. For satisfactory use plans should be drawn to scale and accompanied by sectional elevations. The scale should be large enough to be convenient in producing and so the drawings may be easily read. Locate the building with reference to the compass points and state ceiling heights and the principal dimensions of each room. The beginner will experience some difficulty in the calculations in making proper allowances where absolute values are not obtainable, such as exposures, ceilings, floors, closets and smaller rooms where heat is not provided for. The personal element enters into this part of the work very much and a thorough practical experience is of great value.

In estimating G the simplest and most convenient method is to take it the full area of the sash. That is to say, take the full window opening as glass. Values of K for glass have been quoted from .9 to 1.25 by various authorities. It is the opinion of the author that where the full window opening is used as glass it will be best to make $K = 1$. In Tables VI and XII this value is used. Referring to the Living Room, adding four inches to the width and five inches to the height of each window gives 73×52 and 73×32 inches respectively = 42 square feet total.

Floor registers are shown on the first floor plans but these may be shifted to wall registers if preferred. Tabulations in Table XII show vent registers and ducts in each room. These values may be used for return registers and ducts also. Return lines should be run from each second floor room excepting Bath; also from Study, Dining Room

and Reception Hall on the first floor. Increase the size of the return register in the Hall from 12-in. x 18-in. as calculated to 16-in. x 20-in. and omit the return in the Living Room. Vent registers should be run to the attic from the Bath Room and Kitchen and from such other rooms as desired by the owner.

Where the calculated area of stacks is too great to be included between the studs of a 4-inch wall, a 6-inch wall should be put in. Stacks on the first floor are omitted and where wall registers are used, a floor-wall type is recommended.

The heat line to the Bath Room is a very bad arrangement but is about the best that can be done with the present room plans. To overcome the effects of the cold wall and the resistance of the offset in the floor, set the stack in an offset within the Kitchen and enter and leave the floor horizontal by a good sized turn. Avoid sharp corners.

In selecting the various stacks and leaders it may be well to standardize as much as possible and avoid the extra expense of installing so many sizes. This can be done if the net area is not sacrificed.

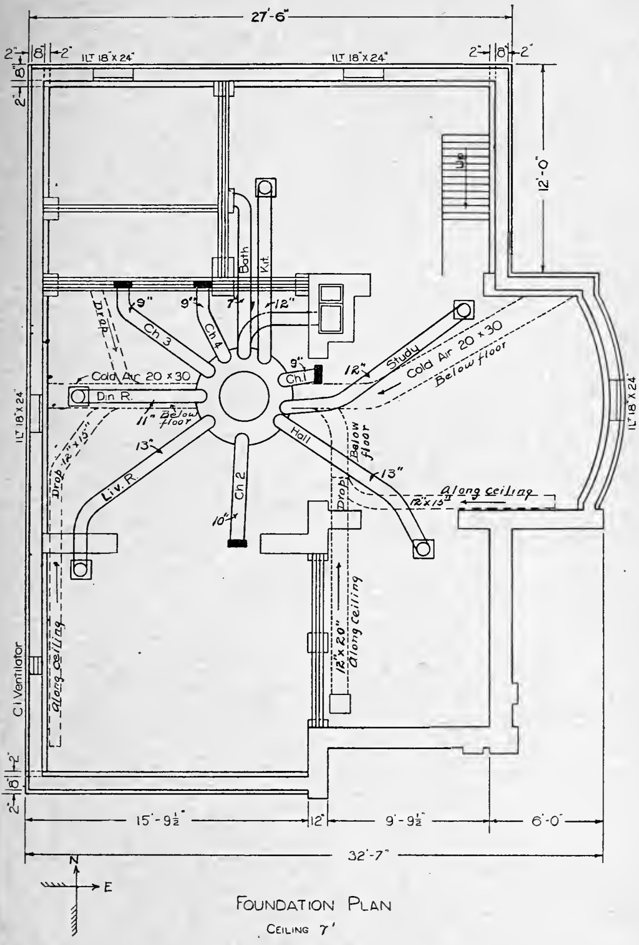
Diameter of grate allowing ventilation for ten people = 26 inches. Cold air duct = 600 square inches = 20 $\times$ 30 inches.

REFERENCES.—*Trans. A. S. H. & V. E.*. Rational Methods Applied to the Design of Warm Air Heating Systems, Vol. XXI, p. 389. Engineering Data for Designing Furnace Heating Systems, Vol. XXI, p. 519.

TABLE XII.

H From Equation 26.

	Living Room	Dining Room	Study	Kitchen	Reception Hall	Chamber 1	Chamber 2	Chamber 3	Chamber 4	Bath	Totals
G	42	32	48	32	16	48	42	32	32	9	333
W	263	114	192	198	390	168	246	103	148	72	1894
F, floor or ceiling	195	-----	-----	138	120	180	194	174	172	72	1245
n	2	2	2	2	3	1	1	1	1	2	-----
C, cu. ft.	1950	2100	1900	1380	1200	1620	1746	1566	936	648	15046
H, B. t. u.	15267	9956	12948	12828	14059	10583	11770	9092	8892	5179	110574
$N. H. R.$, sq. in.	152	99	129	128	140	105	117	90	88	51	-----
$H. R.$, size	14x16	10x14	12x15	12x15	12x18	10x16	12x15	10x14	10x14	8x10	-----
$H. S.$, sq. in.	-----	-----	-----	-----	-----	63	70	54	53	31	-----
Leader, diam.	13	11	12	12	13	9	10	9	9	7	-----
$N. R. R.$ $N. V. R.$ sq. in.	106	69	90	90	140	73	82	63	62	36	-----
$R. R.$ $V. R.$ size	10x16	8x12	10x14	10x14	12x18	9x12	10x12	8x12	8x12	6x10	-----
$R. S.$ $V. S.$ sq. in.	84	55	72	72	78	44	49	38	38	22	-----
REMARKS Basement not ceiled. First floor, floor loss; temperature 40°, $K = .45$. Attic floored solid. Second floor, ceiling loss; temperature 20°, $K = .25$	South exposure. No additions	No additions for exposure or closet because of benefit from furnace below	Add 10 per cent. for exposure	Add one-half east wall to W Add 15 per cent. for pantry and exposure	Take east wall and south wall 19 ft. high. Take volume 10x12x10 and disregard upper hall. Heat and return register same size.	Exposure loss accounted for by benefit from chimney	Closet allowance accounted for by heat from reception hall	Add 10 per cent. for closet and exposure	Add closet wall and glass to room. Add floor over porch same as ceiling. Add 10 per cent. for exposure.	Add 15 per cent. for closet and exposure	



FOUNDATION PLAN
CEILING 7'

Fig. 17.

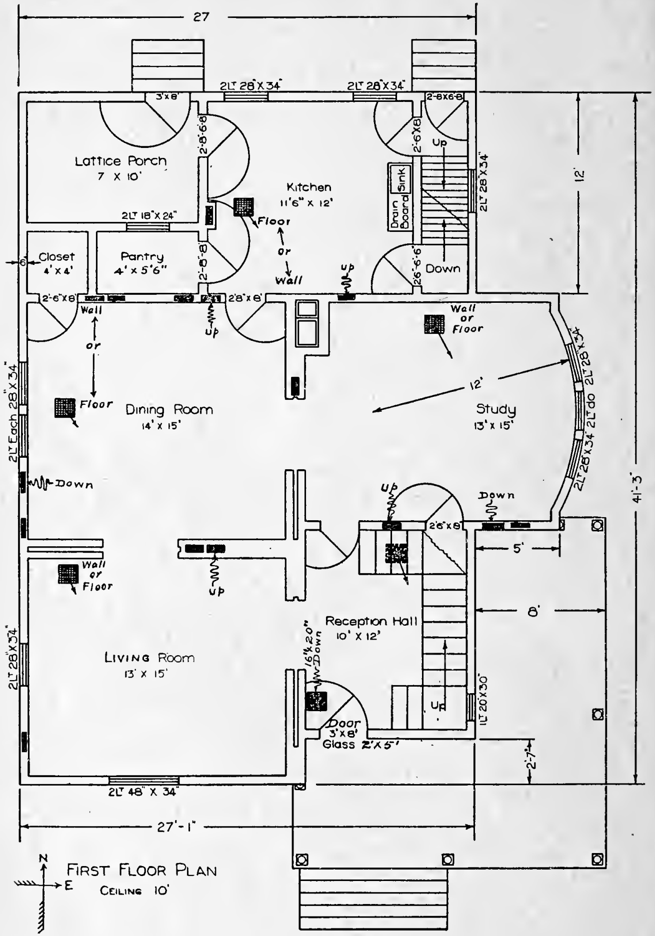

27
2LT 28"X34"
2LT 28"X34"
3'X8'
2-8X6-8
Lattice Porch
7 X 10'
Kitchen
11'6" X 12'
Drain Board
Sink
2LT 18"X 24"
Closet
4' X 4'
Pantry
4' x 5'6"
Floor
or
Wall
Up
Down
2LT 28"X34"
12'
2'-6'X8'
2'8"x 8'
Wall
or
Floor
Wall
or
Floor
Dining Room
14' X 15'
Study
13' X 15'
2LT Each 28"X 34"
2LT 28"X34"
2LT do
2LT 28"X34"
Down
Up
Down
2'6"X8'
5'
8'
Wall
or
Floor
Up
Reception Hall
10' X 12'
16"X20" Down
Living Room
13' X 15'
2LT 28"X34"
1LT 20"X30"
Door
3'X8'
Glass 2'X5'
2'-7"
41'-3"
2LT 48" X 34"
27'-1"
N
E
First Floor Plan
Ceiling 10'

Fig. 18.

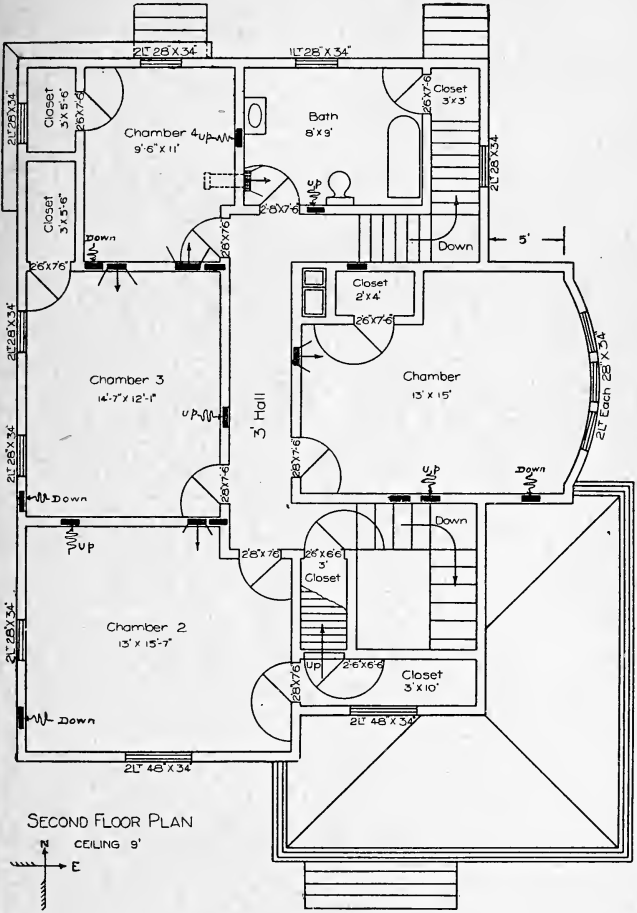

2LT 28"X34"
1LT 28"X34"
2LT 28"X34"
Closet
3'X5'-6"
2'6"X7'-6"
Chamber 4
9'-6"X11'
Up
Bath
8'X9'
Closet
3'X3'
2'6"X7'-6"
2LT 28"X34
Closet
3'X5'-6"
Down
2'8"X7'-6"
2'8"X7'6"
Down
5'
2'6"X7'6"
2LT 28"X34"
Closet
2'X4'
2'6"X7'-6"
Chamber 3
14'-7"X12'-1"
Up
3' Hall
Chamber
13'X15'
2LT Each 28"X34"
2LT 28"X34"
2'8"X7'-6"
2'8"X7'-6"
Up
Down
Down
Up
Down
2'8"X7'6"
2'6"X6'6"
3'
Closet
Chamber 2
13'X15'-7"
2LT 28"X34"
Up
2-6"X6-6"
Closet
3'X10'
2'8"X7'6"
2LT 48"X34"
Down
2LT 48"X34"
SECOND FLOOR PLAN
CEILING 9'
N
E

Fig. 19.

63. Determination of the Best Outside Temperature to Use in Design and the Costs Involved in Heating by Furnaces:—As a basis for the work of the heating and ventilating engineer it is necessary that he be well acquainted with the temperature conditions in the locality where his services are employed. He should compile a chart showing extreme and average temperatures covering a period of years and with this chart a fairly safe estimate may be made upon the costs involved in operating any heating and ventilating system during any part of the average season or throughout the entire heating season. Any estimated costs of operation are only illustrative of method and probability. All one can say is that if the temperature in any one season averages what is shown by the average curve for the period of years investigated, the cost in operating the system may be easily shown by calculation. Heating costs are relative values only and cannot be determined exactly except under test conditions.

The heating engineer should also know the minimum outside temperatures covering a period of years in that locality to determine an outside temperature for his design work. Every design is a compromise between average and extreme conditions, approaching the extreme rather than the average. Patrons expect heating systems to be designed to give normal temperatures in the rooms on all but a few of the coldest days. Extremely low temperature conditions have a duration of from two to three days and it would not be good engineering from an economic standpoint to design the system large enough to heat to normal inside temperature on the coldest day experienced in a period of years. The plant would be too large and would require too much financial input. As an illustration of the method of obtaining the outside temperature to be used in design, also methods of determining approximate costs for heating, see Fig. 20. The low central curve is plotted from the average temperatures on each of the days respectively between September fifteenth and May fifteenth, covering a period of thirty years, at Lincoln, Nebraska. The minimum temperature for December, 1911, and January, 1912, (regarded as a period of unusual severity) are included. Referring to the chart it will be seen that this cold period reached its minimum temperature of —26° on January twelfth. Assuming this curve to represent the most severe weather in this locality, a study

of conditions may easily determine the best outside temperature to be used in design. There were twenty days when the temperature was below zero, twelve days below —5°, six days below —10°, two days below —20°, and part of one day below —25°. Each of the extreme and sudden drops were such as to last from two to three days and were only experienced in two or three instances. It is very evident that a system designed for 0° outside would fall short of the requirement even when put under heavy stress. On the other hand one designed for —25° outside would actually

TEMPERATURE-CHART-AND-HEAT-LOSS-FOR-AVERAGE-YEAR.

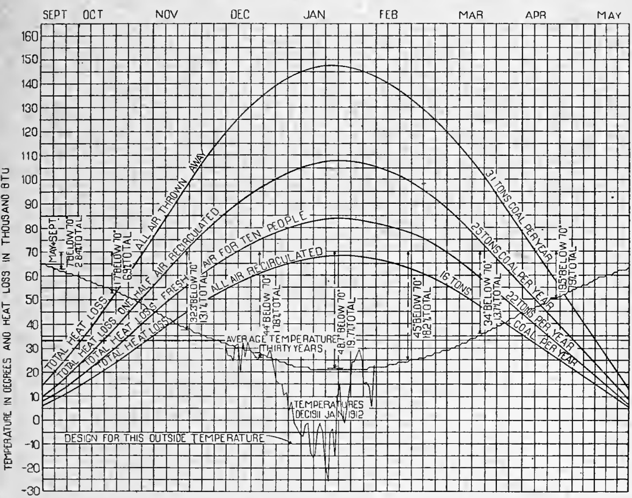


Fig. 20.

come up to its capacity for only a part of one day out of the 240 heating days. One designed for —10° would fulfill conditions without forcing excepting at two or three periods of very short duration, at which times the system could be forced sufficiently without detriment. The personal equation enters into the calculation of the heat loss somewhat and there will be some difference of opinion concerning which to use, —10° or —15°. Probably the latter would be a safer value. All that is necessary is to plan for ample

service at all but one or two of the cold periods of short duration and the system will be considered very satisfactory from the standpoint of size and capacity. Any additional amount put in would be an investment of money which is scarcely justified for the small percentage of time that this additional capacity would be called for.

After the minimum outside temperature has been decided and the plant is designed one would like to know the probable expense in handling such a plant throughout the heating season. Assume an inside temperature *throughout the building* of 70°. Combine the two half months, September and May, into one month, and take the average of these average temperatures for the days of each month, thus giving the drop in temperature between the inside and the outside of the building. The heat loss from the building is approximately proportional to these drops in temperature. In this case the differences are as follows:

September + May	7°	below 70°
October	17°	below 70°
November	32.3°	below 70°
December	44°	below 70°
January	48.7°	below 70°
February	45°	below 70°
March	34°	below 70°
April	19.5°	below 70°

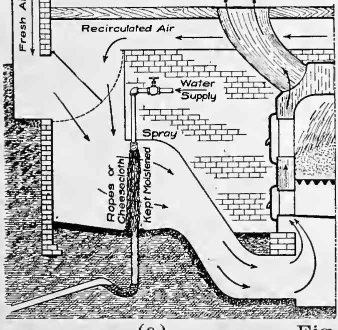
Taking the sum of all these differences as the total, 100 per cent., and dividing each individual difference by the total, we have the percentages of loss for the various months as follows:

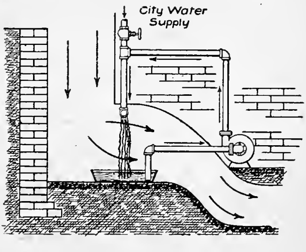
September + May	2.84	per cent. of total yearly loss
October	6.86	" " " " "
November	13.05	" " " " "
December	17.77	" " " " "
January	19.67	" " " " "
February	18.20	" " " " "
March	13.71	" " " " "
April	7.90	" " " " "
Total	100.00	

These percentages of loss indicate what may be expected in the expense for coal for the respective months of the average heating year in the locality stated. Upon this

basis, Fig. 20 represents an application of the above to a residence having a heat loss, H, approximating 100000 B. t. u. per hour. The results are shown in B. t. u. loss and in tons of coal per year, assuming that the entire house is heated to 70° upon the inside for each hour between September fifteenth and May fifteenth. The lowest curve is that for direct radiation only. The next superimposed curve assumes outside air for ten people. The third curve assumes one-half of the required air to be recirculated and the upper curve assumes all the air to be from the outside.

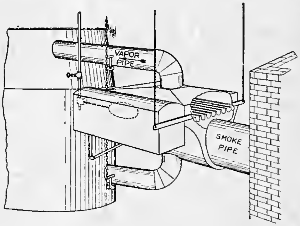
64. Filtering, Washing and Humidifying Furnace Air:— Two objections frequently urged against indirect heating are the dust content and the dryness of the circulated air. It is possible to overcome both of these objections in the larger mechanical plants, but in small furnace plants where the circulated air moves wholly by convection but little progress has been made. Cheesecloth screens, fibrous material such as unbraided ropes, linen strips and soft wicking, kept moist by water drips, may be used for filtering. To offset the air resistance due to the filter, the cross sectional area of the duct at the filter should be at least four times the net area of the duct, for cheesecloth and fibrous material, and twice the net area for linen strips and wicking. The filter should be frequently cleaned by flushing. Where water pressure from city supply or pump is available, *sprays are more satisfactory than the filter.* The essentials of a small residence air washing plant are: pressure water supply, electric current, washing chamber and drainage. Fig. 21, *a* shows

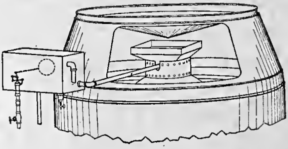




(a) Fig. 21. (b)

a simple arrangement. Having given a furnace heating plant, provide an air mixing chamber for the fresh and recirculated air, install a spray head (a $\frac{1}{4}$- or $\frac{3}{8}$-inch pipe built in the form of an ell or tee, with the ends plugged and the bottom drilled with $\frac{1}{64}$-inch holes), from this spray head suspend unbraided ropes, hemp strands, linen strips, wicking or layers of cheesecloth kept wet by water drips and compel the air to weave its way through these wetted surfaces to the furnace. Much of the dust and other mechanically suspended particles in the air will be deposited on the fibrous material and finally washed to the sewer. Because of the low pressure head moving the air in such a plant *no unnecessary resistances should be put in.* The chief objection to the system shown is the water waste which may be any amount, from just enough drips to keep the eliminators moist, to the full jet outlet under pressure. Water waste may be almost wholly eliminated by catching it in a metal basin and recirculating it by means of a small electric pump, as in Fig. 21, *b*. But here again is the expense of operating the electric motor and the cost of the small amount of water that is thrown away when flushing and refilling. The operating expense in any of these systems is not excessive as shown under the application. Where air washing is not considered, humidity conditions may be cared for by evaporating pans as suggested by Figs. 22, (*a*) and (*b*).





(a) Fig. 22. (b)

Application.—A ten-room residence circulating not to exceed 100000 cubic feet of air per hour has a fresh air duct (or recirculating duct) 4.2 square feet in cross-sectional area. Because of the resistance offered to the circulating air let the area be 16 square feet at the filter, say 4-ft. x 4-ft. Let the spray head be a single line 4 feet long with ten, $\frac{1}{64}$-inch holes on the under side. What will it cost to operate such a washing plant for a day of 15 hours under maximum

water flow by each of the two systems, i. e., the waste water system, where the water is run to the sewer continuously, and the recirculating system where the only loss is that due to occasional flushing. City water may be assumed to have a pressure of 50 pounds per square inch and to cost 15 cents per 1000 gallons. Electric current may be had for 10 cents per K. W. At a gage pressure of 50 pounds, a $\frac{1}{64}$-inch hole will jet approximately one cubic foot of water (62.5 lbs.) per hour. Ten holes, therefore, will waste to the sewer 625 pounds (75 gals.) per hour. This water will cost $15 \times \frac{75}{1000} =$ 1.125 cents per hour, or 16.8 cents for a 15-hour day.

The electric motor pump may be made to circulate a sufficient amount of water without waste, at a pressure much less than 50 pounds. For comparative values in calculation call the pressures the same. With an efficiency of the motor pump 40 per cent., the work done per hour by the motor is $(625 \times 50 \times 2.3 \times .746) \div (33000 \times 60 \times .40) =$ .068 K. W. At ten cents per K. W. this is .68 cent per hour or 10.2 cents for a 15-hour day. No allowance is here made for the small amount of friction loss in the nozzles and pipes as these vary greatly with the amount of pipe and the pressure under which the system is run. The amount lost in evaporation is the same in each case.

In the two estimated values, that for the waste water system would probably decrease in the average plant because of a saving of water by throttling the supply, while that for the pumping plant would probably increase slightly. A very fair estimate in either case is *one cent per hour* for maximum service. This is sufficiently large to allow for a material increase in the number of the jets above that given.

Where an electric motor pump is used to circulate the water or where the city supply is used without throttling, the filters may be omitted, the duct may be uniform in section and the spray head may be located across the bottom of the opening with the holes pointing toward the top of the duct. In this way the spray is broken up by contact with the deflector at the top of the duct and falls as a mist through the air current.

It would be of interest here to briefly discuss the *probable temperature and humidity effects* upon the circulating air within the residence if a washing plant of this character were installed. This section is offered as a fair probability

in the absence of collected data. For basis of argument, assume the following: 100000 cubic feet or air circulated per hour at the register, register temperature 120°, no recirculated air used, temperature outside 50°, temperature inside 70°, humidity outside 50 per cent., humidity inside 45 per cent. What amount of water is absorbed per hour?

100000 cubic feet of air at 120° at the register is equivalent to 87930 cubic feet at 50° on the outside and 91380 cubic feet at 70° in the room. The amount of moisture in a cubic foot of saturated air on the outside at 50° is 4 grains and at 50 per cent. saturation would be $.50 \times 4 = 2$ grains. Correspondingly in the room air at saturation we have 7.98 grains and at 45 per cent. humidity $.45 \times 7.98 = 3.59$ grains. The total amount of moisture in the incoming air is $87930 \times 2 = 175860$ grains (25.12 lbs.) per hour. The total amount of moisture in the room air is $91380 \times 3.59 = 328054$ grains (46.8 lbs.) per hour. It is evident that the difference between these two amounts ($46.8 - 25.12 = 21.73$ lbs.) has been added per hour from the washing water (See Art. 27). In this way the weight of water absorbed may be worked out *theoretically* for any temperatures and for any humidities. The *actual* amounts absorbed, however, may vary considerably from the theoretical figures because of the wide range in temperatures and humidities between the incoming and outgoing air and the shortness of time the air is in actual contact with the water.

Close *regulation of the humidity* of such a plant is a difficult problem. The humidostat, if used, necessarily acts to control the amount of water flowing. When the humidity is high this cuts off the flow of water, in which case the apparatus ceases to serve as a washer. From what is known of such plants it is probable that the humidity of the air after passing the furnace is never high enough to give much concern and the humidostat may be eliminated. The location of the spray head, in the cool air chamber, retards the absorption process because cool air takes up moisture with less freedom than warm air. Even assuming that the cool air is fully saturated as it enters the furnace, the humidity will drop so rapidly as the air is heated that there will never be any danger of depositing moisture on the furniture of the room. To illustrate.—In the above problem assume that the 50° air is saturated as it enters the furnace (a condition it will seldom reach). When heated

to 70° the humidity will be 52 per cent., which is a very satisfactory amount. Again, if the outside air is 60° and saturated as it enters the furnace, the humidity, when raised to 70° will be 75 per cent., an amount that would still cause the air to be agreeable. Now what happens at low temperatures? If the entering air is saturated at 40°, the humidity at 70° would be 37 per cent. From this it would seem that a humidostat, for the purpose of controlling the moisture, would be of very little service unless the air were circulated for purposes of ventilation at or near 70° and saturation, a state of affairs very seldom asked for in residence work.

CHAPTER V.

FURNACE HEATING AND VENTILATING.

SUGGESTIONS ON THE SELECTION AND INSTALLATION OF FURNACE HEATING PARTS.

65. The Furnace:—Furnaces for residences are usually of the portable type, illustrated by Fig. 23. This consists of a heating stove enclosed in a shell composed of two metal casings having a dead air space or an asbestos insulation between them. Some of the larger furnaces used in the larger residences, small schools, etc., have permanent casements of brick work as in Fig. 24. Both types of furnaces give

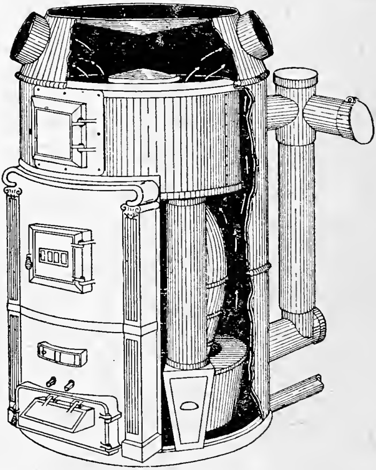


Fig. 23.

good results. The points usually governing the selection between portable and permanent settings are required capacity, price and available floor space.

The stoves are made of cast iron, wrought iron and steel. The cast stove admits of a greater variety of shapes than those made of rolled plates hence it is more commonly

used. The sections are assembled with cemented joints while the rolled sections are riveted. The chief objection to the cast stove is the frequent leakage of fuel gases from the combustion chamber to the warm air passages. In a properly designed and set up cast furnace there should be little excuse for leakage. When it does occur, examine the cement in all the joints especially around the door openings. It is claimed by some that the heated cast iron plates permit the passage of some of the gases directly through the metal. While this may be true to a certain degree in comparison with rolled materials such as steel, there is little doubt that practically all leakage can be traced to cracked sections or to broken cement joints. In general, the fewer the joints

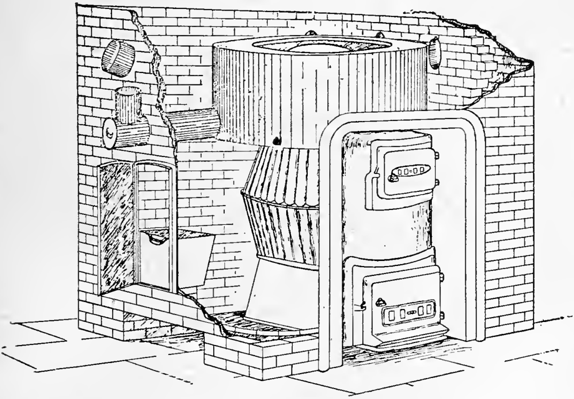


Fig. 24.

in a furnace stove the better. On the other hand cast iron corrodes less than rolled plate and the heavy cast walls of this type act as a storage for heat and tend toward less fluctuation of air temperature.

Furnaces are *direct-draft* and *indirect-draft*. In the direct-draft type the radiator (heat distributor) is above the fire and the gas passages are usually short and fairly direct to the chimney. In the indirect-draft type the radiator is below the fire and the gases are first deflected downward over the radiator and then upward to the chimney. In this type there should always be a by-pass, properly dampered, so that when there is a lack of draft due to a cold chimney

(usually found when starting a new fire) or other cause the gases may be given a short cut to the chimney removing excess friction and avoiding smoking. An indirect-draft furnace should be used only on a protected or enclosed chimney. In cases where sufficient draft is sure at all times this type of furnace is probably the most economical.

The cylindrical fire pot is better than a conical or spherical one, there being less danger of the fire clogging and becoming dirty. A lined fire pot is better than an unlined one, because a hotter fire may be maintained in it with less detriment to the furnace and less contamination of the air supply. There is a loss of heating surface in the lined pot, however, and in most furnaces the fire pot is unlined to obtain this increased heating surface.

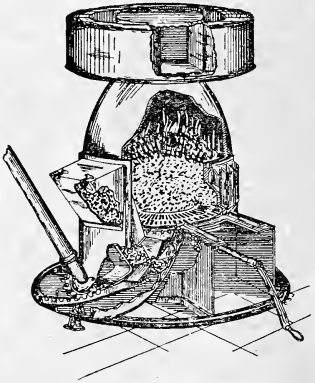


Fig. 25.

Some form of shaking or dumping grate should be selected, as a stationary grate is far from satisfactory. Care should be exercised in the selection of the movable grate, as some forms not only stir up the fire but permit much of it to fall through to waste when being operated.

In most furnaces the fuel is fed to the fire pot from a door above the fire. These are called *top-feed* furnaces. In some forms the fuel is fed up through the center of a rotary ring grate. These are called *under-feed* furnaces (Fig. 25) and for the finer grades of coal are preferred to the top-feed furnaces.

The *size of the furnace* for any given work may be obtained from the estimated heating capacity in cubic feet of

room space as given in Table 20, Appendix. A better and safer way, and one that serves as a good check on the above, is to select the furnace from the *calculated grate area* (See Art. 60).

A *combination furnace and heater* is shown in Fig. 26. With it some of the rooms of a residence may be heated by warm air and the remainder by hot water or steam. In this way rooms to be ventilated as well as heated may be connected by the proper stacks and leaders to the warm air deliveries, while rooms requiring less ventilation or heat only, or those rooms that are difficult to heat with air circulation may have radiators installed and connected to the flow and return pipes of the water or steam system.

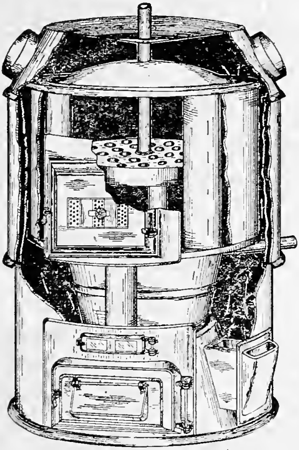


Fig. 26.

Pipeless furnaces are manufactured and installed in stores, small residences and the like. In this type the stove is surrounded by two independent casings instead of one as in the pipe furnace, heated air circulating upward between the stove and the inner casing and return air downward between the casings. The top of the furnace terminates in a short stack capped by a combination hot-and-cold air register at the floor line of the first floor room. The central zone of the register supplies air to the room and the outer ring zone

carries cold air from the room to the furnace. All air is delivered to one room and from here circulates to and from the other rooms of the house through transoms, open doors, etc. Fig. 27 shows the principle of operation. Compared with other furnace plants the application of this type is

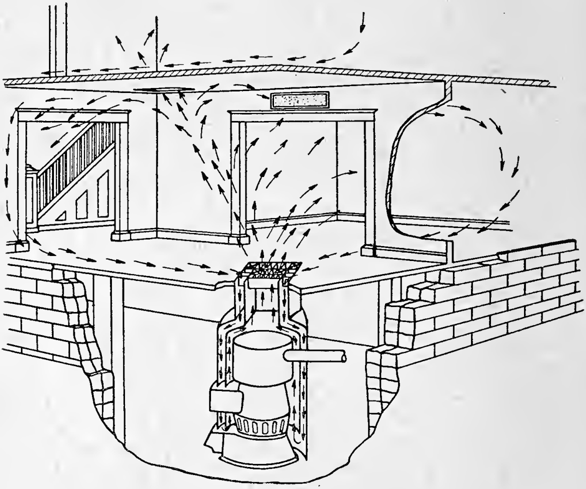


Fig. 27.

much simpler and the installation cost is less. Its satisfactory application is limited to those buildings having open interior construction where the furnace may be centrally located and where every room has continuous opening to the room above the furnace. Furnaces of this type are good heaters but have no ventilating possibilities. One of the objections usually found with this type of furnace is the presence of floor drafts in the room above the furnace. Since bath rooms, toilet rooms, laundries and kitchens are usually not connected to the return circulation, these rooms are difficult to heat from the pipeless furnace.

Room Heaters (slightly modified forms of standard furnaces) may be obtained for use in small buildings having no basements. Such furnaces should have well insulated metal jackets to protect the nearby room furnishings from excessive heat. The circulating air may be taken from the out-

side of the building in about the same way as a direct-indirect radiator (See Art. 102), or may be recirculated from the room, entering the bottom of the furnace at the floor line through a register base, and leaving from the wide open top of the furnace. A vent flue for the room is usually provided by the side of or in connection with the chimney or smoke flue. Room heaters are naturally not desired because of their appearance but they are effective heaters and where properly installed may give fair ventilating effect. The one serious objection to the room heater, other than its appearance, is the presence of floor drafts as in the pipeless furnace.

One of the most important points in the selection of any furnace is its *cleaning* possibilities. If there is a probability that soft coal may be used at any time the furnace should be provided with clean-outs so situated that all parts of the gas passages may be reached by flue swabs.

Care must be exercised also in the installation of the furnace to protect the nearby combustible material from fire. Smoke pipes, especially, must have ventilated thimbles giving at least 2-inch air space all around the pipe where passing through combustible walls.

66. Location and Setting of the Furnace:—A furnace should be set as near the center of the house plan as possible. Where this can not be done, preference should be given to the colder sides (sides subjected to the heaviest winter winds), in most localities the north and west. In any case, it is advisable to have the leader pipes of uniform length and pitch if possible. The smoke pipe should be short, but it is better to have a moderately long smoke pipe and obtain a more uniform length of leaders than to have a short smoke pipe and leaders of widely different lengths.

The furnace should be set low enough to give a good upward slope to the leaders from the furnace to the respective stacks. This should be *not less than one inch per foot of length and more if possible.* Each leader should be dampered near the furnace.

The location of the furnace will call forth the best judgment of the designer, since a right or wrong decision here is very vital.

Foundation.—All furnaces should have the manufacturer's directions to govern the setting. Such information is

usually followed. In every case the furnace should be mounted on a level, brick or concrete foundation especially prepared and well finished with cement mortar on the inside, since this interior is in contact with the fresh air supply.

67. Fresh Air Duct:—Ducts below the floor are best constructed of hard burned brick walls 4 inches thick, concrete walls 2 to 3 inches thick or vitrified tile; the floors to be not less than 1-inch concrete and the tops to be 1- to 2-inch concrete slabs. The walls and floors of the brick or concrete ducts should be smooth plastered with neat cement and all joints should be tight.

Ducts above the floor are usually made of galvanized iron. Where made of boards they should be solid material well tongued and grooved. The riser from the main horizontal to the outside of the building may be of wood, tile or galvanized iron and the fresh air inlet should be vertically screened. The whole fresh air line should have tight joints and should be so constructed as to be free from surface drainage, dirt, rats and other vermin.

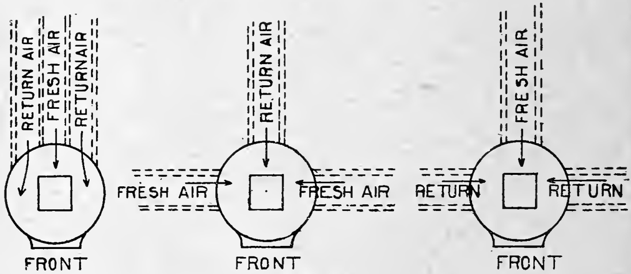


Fig. 28.

In addition to the opening for the admission of the fresh air duct, another opening or openings may be made under the furnace for the purpose of admitting the duct which carries the recirculated air from the rooms to the furnace (See Fig. 28). Occasionally the two ducts unite in a Y fitting before entering the furnace, in which case the fitting should be so constructed as to make the two uniting streams of air enter as nearly parallel as possible. Each of these ducts should have adjustable dampers so as to make them independent of the other. Each duct also should be provided with a door that can be opened temporarily to the basement for inspection and cleaning. Sometimes it is de-

sirable to have two or more fresh air ducts leading from the different sides of the house to get the benefit of any change in air pressure on the outside of the building (See also Figs. 16 and 17).

Arrangements may be made for pans of clear water in the air duct entering the furnace (See Art. 64) to give moisture to the air current, but it should be understood that only a small amount of moisture will be taken up at this point from still water surfaces. In most cases where moistening pans (commonly called *water pans*) are used, they are installed in connection with the furnace itself. Furnaces should have special means provided for moistening the circulated air. The water pan is a step in the right direction, but this alone is not sufficient (See Art. 27).

68. Recirculating Ducts:—Ducts should be provided from the rooms within the building, through the basement to the bottom of the furnace. These ducts carry the air from the rooms back to the furnace to be reheated for use again within the building. *Recirculating the air gives a more positive type of heating system.* Rooms difficult to heat without recirculation are improved with its use and rooms at a distance from the furnace should always have it. Frequently a number of rooms are grouped together on one return line. Small residences may have but one return line leading from the return register in the front hall near the door. Return lines should be grouped in the basement to simplify the system and to avoid making many openings into the furnace foundation. Return stacks should be light tin or galvanized iron built-in between the studding of the outside walls and need not be insulated. Contractors usually omit these metal ducts between the studs, and the dust from the rooms settling on the rough surfaces of the studs and sheathing makes an unsanitary condition. In like manner horizontals in the basement are frequently slighted by tinning under two adjoining joists thus forming the duct. Such construc-

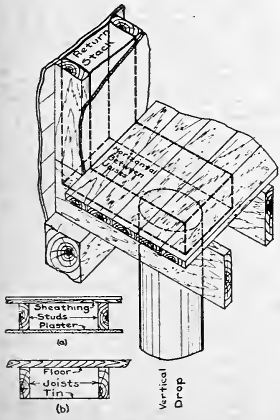


Fig. 29.

tion should not be permitted. *All vertical return lines and all exposed horizontal runs between the return registers in the rooms and the attachment at the furnace, should be tin or galvanized iron with tight joints.* (See Fig. 29). Avoid overhead return ducts near the furnace. In some installations it is necessary to carry these ducts along the basement ceiling part way across the basement but the drop to the floor should be made at such a distance from the furnace that the air in the vertical return will not be retarded in its fall by the heat from the furnace.

69. Leaders:—All leaders should be round and free from unnecessary turns. They should be made from No. 24 or No. 26 galvanized iron or tin, should be run as straight as possible and should be well supported. Connections with

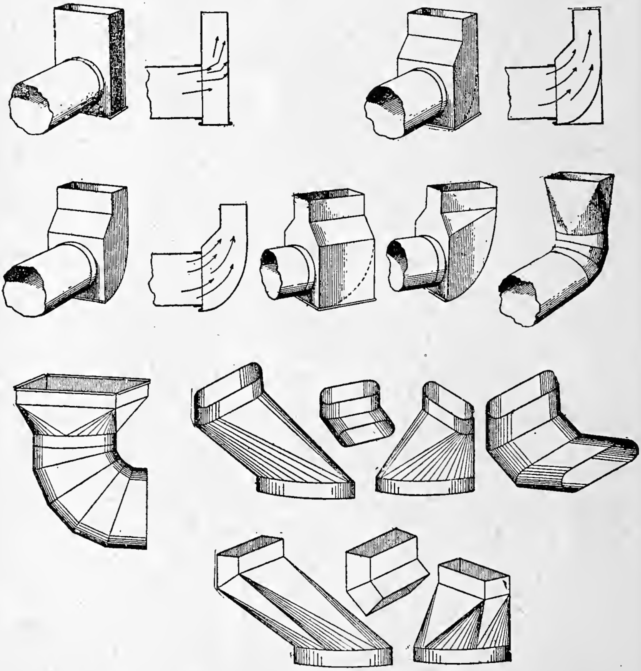


Fig. 30.

the furnace should be straight, but if a turn is necessary, provide long radius elbows. All connections to risers or

stacks should be made through long radius elbows. Rectangular shaped *boots* having attached collars are frequently used but these are not satisfactory because of the impingement of the air against the flat side of the stack; also, because of the danger of the leader entering too far into the stack and shutting off the draft. Leaders should connect to first floor registers by long radius elbows. Leaders should have as few joints as possible and these should be made firm and air tight. Fig. 30 shows different methods of connecting between leaders and stacks, and between leaders and registers.

Leader pipes should be covered to avoid heat loss and to provide additional safety to the plant. The covering usually put on is one or two thicknesses of asbestos paper laid with face contact. As a heat insulator this is little better than the bare pipe. A better way is to have the layers of asbestos paper separated by spiral wrappings of wire, air cell material or mineral wool to give dead air spaces. Leaders passing through combustible walls must have ventilated thimbles, giving at least a 1-inch air space all around the leader.

70. Register Connections:—The most efficient first floor

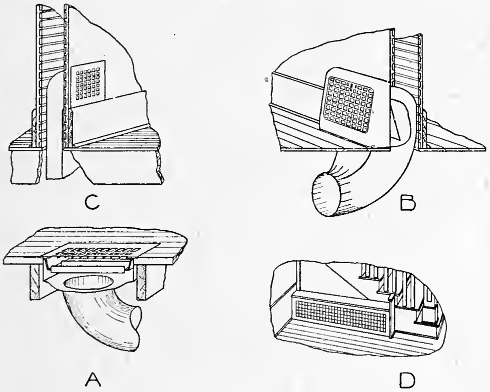


Fig. 31.

warm air intake is through a *floor register*. Fig. 31 *a* shows a galvanized floor box enclosing a floor register and connected to the leader by a round elbow. Floor registers give

the freest circulation that can be obtained in first floor furnace heating, but they are dust catchers and unsanitary; also, in rooms having hard wood floors and special furnishings they are not usually permitted. In such cases the *floor-wall register* may be used as in Fig. 31 *b*. This type has a box and leader opening much larger than is possible with a wall stack and compares favorably with the floor type. The appearance of the floor-wall register as a feature of the room furnishings and its sanitary qualities are enough better than the floor type to justify its general use. On the second floor a stack is necessary and the *wall register* is generally used (Fig. 31 *c*). Where these are installed the upper end of the stack should terminate in a quarter turn to throw the warm air toward the room and avoid eddy currents at the dead end.

All intake registers to the rooms and vent registers leading to the attic should be provided with shutters. Return lines may have register faces only. For large register faces where strength is not an important factor, grilles made from latticed wood strips are frequently used. Fig. 31 *d* shows one of these under a hall seat. For calculations and sizes of registers see Arts. 53-55, and Tables 19 and 21, Appendix.

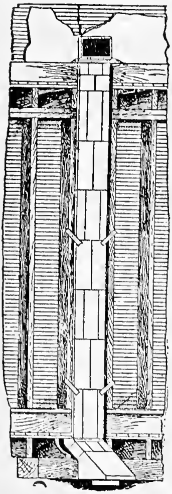


Fig. 32.

71. Stacks or Risers:—The vertical air pipes leading to the registers are called *stacks* or *risers*. They are rectangular in section and are usually fitted within the wall (See Fig. 32). The size of the studding and the distances these are set, center to center, limit the effective area of the stack. All stacks should be insulated to protect the woodwork. This is done by making the stack small enough to clear the woodwork by at least ¼-inch and then wrapping it with some nonconducting material such as asbestos paper or hair felt held in place by wire. Patented *double walled stacks* having an insulating air space between the walls are more nearly fireproof. All stacks should have tight joints and should have ears or flaps for fastening to the studding. Patented stacks are made in standard sizes and of various lengths. Sizes ordinarily used in practice are given in Table 17, Appendix.

A stack is sometimes run up in a corner or in some recess in the wall of a room where its appearance, after being finished in color to compare with that of the room, is not unsightly. This is necessary in any case where the stack is installed after the building is finished. It is also preferred by some because of its additional safety and because more stack area may be obtained than is possible when placed within a thin wall.

All stacks should be located in or near partition walls looking toward the outside or cold side of the room. This location protects the air current from excessive loss of heat, as would be the case if placed in the outside walls. It also provides a more uniform distribution of the air from the furnace.

The area of the stack best adapted to any given room should not be taken by guess (See Art. 56). In a great many cases the architect specifies light partition walls between large upper rooms, say 4-inch studding set 16-inch centers between 12-foot by 15-foot rooms, heavily exposed. From the theoretical calculations of heat losses, these rooms require larger stacks than can be placed between studding as stated, but it is very common to find such rooms provided for in this way. One possible excuse for such practice may be the fact that most second floor residence rooms are designed for sleeping rooms and not for living rooms. Regardless of this fact, however, it would be well to provide for all emergencies and follow the rule that *every room should be provided with facilities for heat as if it were to be used as a living room in the coldest weather.* If this were done there would be fewer complaints of defective heating plants and less migrating from one side of the house to the other on cold days.

Lack of heating capacity for any one room may be overcome by providing two stacks and registers instead of one. This plan will be fairly satisfactory since one of the registers may be shut off in moderate weather. It requires an additional expense, however, which is not justified. A better way is to provide partition walls of greater thickness or specially planned-for stack openings, so that ample stack area may be put in. The ideal conditions will be reached when the architect anticipates the heating requirements and provides air shafts of sufficient size to accommodate round or nearly square stacks.

Single stacks are sometimes used to supply air to two adjoining rooms. Such stacks have metal partitions extending down a few feet from the upper end to split the air current and direct it to the rooms. This practice is questionable because of the liability of the pressure of air in the room on the cold side of the house forcing the heated air around the partition to the other room. Also, single stacks are frequently used to supply rooms one above the other. This is not satisfactory except where the regulation in each room is taken care of by the same person. When the upper register is full open it will rob the lower register and when the lower damper is full open the upper room gets no heat. A better method is to *install a separate line for each room to be heated.*

Vent stacks should be located in the inner or partition walls and should lead to the attic. If it is thought necessary, they may there be gathered together in one duct leading to a vent through the roof. It is an ideal arrangement but not always necessary to have a vent stack in every room. Some rooms, from their location, are easily ventilated without them. *Bath rooms, toilets, laundries and kitchens, and rooms near the center of the house should always have independent ventilation.* In any rooms where natural ventilation is an important feature, vent ducts to the attic should have two tappings, floor and ceiling, each provided with shuttered registers. The floor vent should be used on cold days to economize the heat and the ceiling vent should be used on warm days. Such a system of ventilation may be used in connection with direct-indirect radiators in small schools (See Fig. 33).

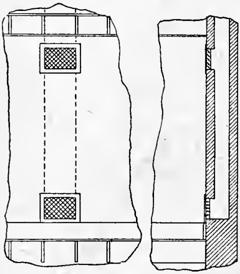
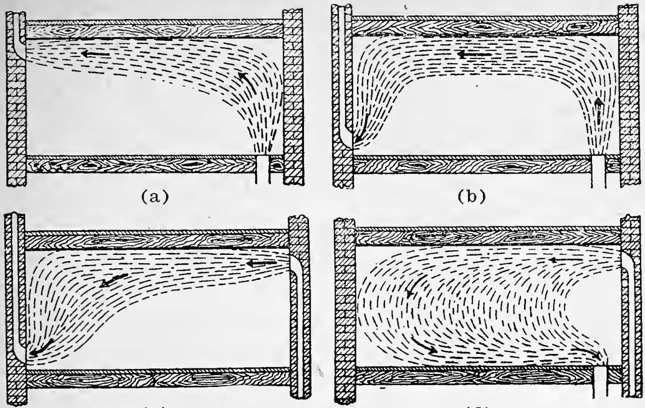


Fig. 33.

72. Air Circulation Within the Room:—The location of the heat register relative to the vent register, will determine to a great extent the circulation of air within the room. Fig. 34 *a*, *b*, *c*, and *d*, shows the effect of the different locations in forced circulation. The best plan, from the standpoint of heating, is to enter the air at a point above the heads of the occupants and withdraw it from the floor line, at or near the same side from which the air enters. This



(a) (b) (c) (d)

Fig. 34.

gives a more uniform distribution as shown by the last figure. It is doubtful, however, if this method will give the best ventilation in crowded rooms where the foul air naturally collects at the top of the room. Circulation in furnace heating is not as satisfactory as in other forms of indirect heating. Air usually enters the room from the floor or the inner wall near the floor line and leaves for recirculation near the floor line on the opposite or cold side. Circulation within the room is shown by *b*. Where there is no recirculation it leaves through a vent register usually near the floor line and located at such a point that the air will traverse as much of the room as possible before leaving.

73. Fan-Furnace Heating System:—In large furnace installations where the air is carried in long ducts that are nearly, if not quite horizontal and where a positive supply of air is a necessity in all parts of the building, a combination fan and furnace system may be installed. Such systems may be properly designated mechanical warm air systems but they should not be confused with the mechanical fan-coil systems described in Chapters X to XII. The objections urged against the fan-furnace systems are the high temperatures of the circulating air and the smoke and dust content picked up from the furnace.

Fan-furnace systems may be set in multiple if desired, i. e., one fan operating in connection with two or more fur-

naces. Fig. 35 represents a two-furnace plant showing a fan and two furnaces. Air is drawn into the fresh air room through a grate in the outside wall and is forced through the fan to the furnaces where it divides and passes up through each furnace to the warm air ducts. Part of the fresh air from the fan is by-passed over the top of the furnaces and is admitted to the warm air ducts through mixing

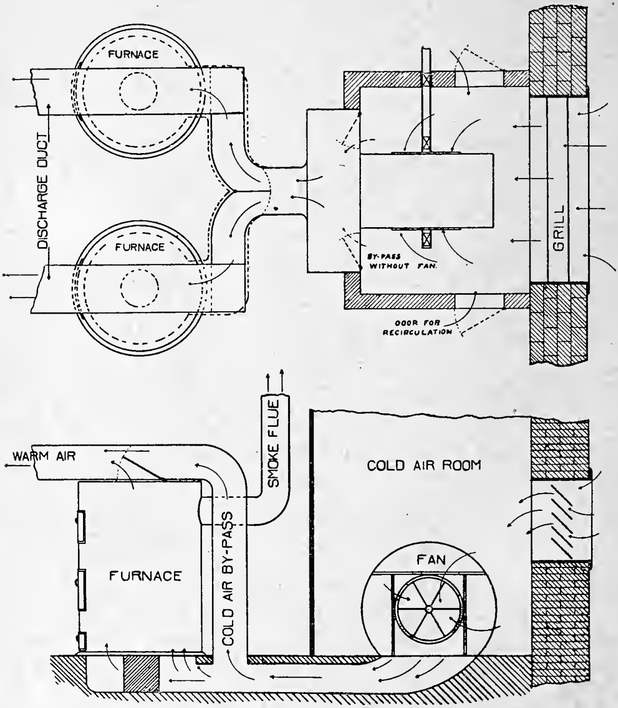


Fig. 35.

dampers. These dampers control the amount of hot and cold air for any desired temperature of the mixture. Temperature control may be installed and operated with this system. Paddle wheel fans (always located between the furnace and air intake) are preferred, although the disk wheel may be used where the pipes are large and where the air must be

carried but short distances. For fan types see Chapter X.

74. Hot Air Radiator Systems:—In some localities gas, either natural or manufactured, is used as fuel for heating purposes. Wherever the supply is available at rates commercially reasonable, it may be piped direct to radiators within the rooms and burned as in ordinary gas stoves, the products of combustion being circulated through the radiators and then exhausted. This is the principle of the *Rector system* (Fig. 36). The gas supply to each burner is under double control: first, by a thermostat which maintains constant room temperature; second, by vacuum produced by the exhaust fan which acts as a safety appliance. For thermo-

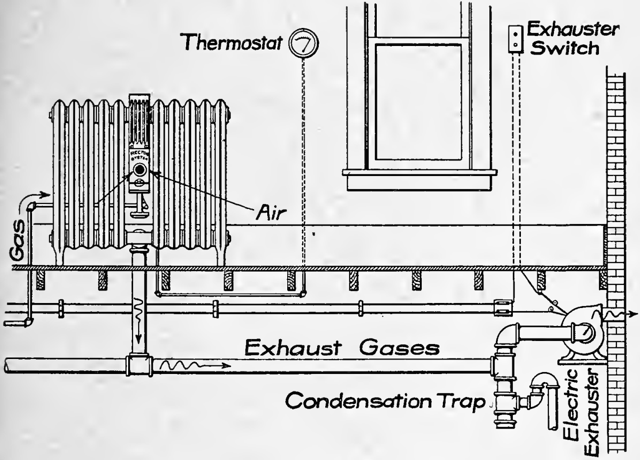


Fig. 36.

static control see Chapter XIV. For vacuum control, a valve in each radiator is so arranged that when the vacuum fails, due to the stopping of the fan, gas is shut off from the burner, leaving only a pilot light flame. When the vacuum is again produced by the starting of the fan, gas is admitted to the burner and ignited automatically by the pilot light. The products of combustion pass upward to the top and then downward through the sections where they are drawn off from the bottom central connection by the exhaust fan. No water, steam or circulating mediums other than the combustion products themselves, are used. The facts that combustion

takes place within the room to be heated, that the only loss of heat is that carried off by the exhausting gases and that each room is an independent unit give just claim for high efficiency in fuel economy. Such a system is practically under the control of a push button which starts and stops the motor exhaust fan. It is very convenient with its flexibility and independence of units and is conducive to economy under careful management. The calculated radiator surface is less than that required for steam systems, because of the higher average internal temperature.

Gas radiators should not be placed close to woodwork or other inflammable material. In general, advantages such as no janitor service, no coal storage spaces, no furnace chimney, no coal dust and dirt, and no ashes are inherent with these systems. They are used in localities having mild climates and where continuous firing is not necessary.

In the *Hawkes system* the exhaust fan and the automatic gas valve of the Rector system are eliminated. The products of combustion pass through radiators similar to those just

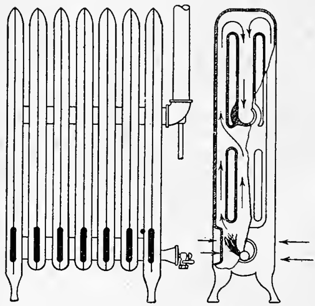


Fig. 37.

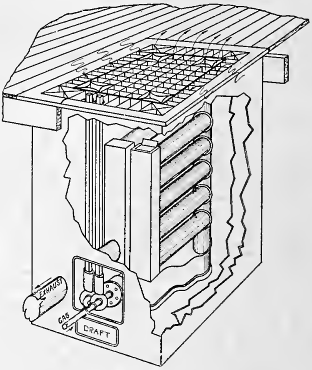


Fig. 38.

mentioned but the exhausting of these products is accomplished by connecting each radiator to a stack, or by providing a separate 2-inch riser to the roof which acts as a stack for that particular radiator. The air necessary for burning the gas is admitted through slots near the bottom of each section of the radiator. All radiators are operated by hand control in the same way as the ordinary gas stove. Fig. 37 shows the Hawkes ventilating gas radiator as commonly installed.

Hot air radiators heated by gas may also be of the indirect type in which case they are designated *gas floor furnaces.* Fig. 38 shows one of these furnaces connected to a first floor register. The operation is like that of the pipeless furnace, Art. 65. Above the furnace is a combination hot-and-cold air register which recirculates the room air over a gas heated cast radiator. Combustion takes place within the cast radiator and the gases are carried by vent pipes to the chimney. These furnaces are hand controlled at the register.

75. Fire Hazard:—Protection against fire from heating apparatus is too little considered by the average householder. Several points in every furnace plant may be considered danger points. These are, in the order of importance: a loosely built chimney, the top of the chimney too close to a steep pitched shingle roof, wood work of the house fixed rigidly to the chimney, the smoke pipe from the furnace too close to the house framing, the top of the furnace unprotected and too close to the joists or basement ceiling, and the hot air pipes too close to the wood work. Especial care should be taken in protecting gas floor-heaters, as described in Art. 74. The ounce of prevention in such cases may be easily and cheaply applied, and should be insisted upon.

76. Accelerating Circulation in Furnace Plants:—Many furnace plants are not giving satisfaction because of sluggish circulation. This trouble which in most cases may be traced to defective design, may be corrected as in Fig. 39, by inserting a 12- to 16-inch disk fan in the return duct, preferably below the inlet point of the outside air. The fan may be run when warming up the house in the mornings and at times of severe weather. This may be connected to the average lamp socket and will cost from ½ to 1 cent per hour depending upon the electric prices in the locality.

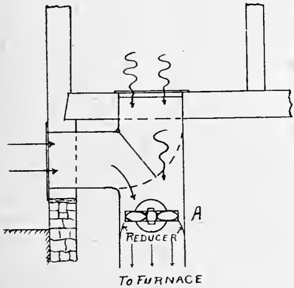


Fig. 39.

77. Suggestions for Operating Furnaces:—Furnaces are designated *hard coal* and *soft coal*, depending upon the type of design and the construction of the grate, hence the grade of coal best adapted to the furnace should be used. The size of the openings in the grate should determine the size of coal used.

Keep coal in coal pile moist but not wet.

Clean all furnace gas passages frequently.

Keep the fire pot well filled with coal and have it evenly distributed over the grate, firing light and often for best service. In a properly designed plant, when necessary, firings may be as few as three or four per 24 hours and give good service.

Keep the fire free from clinkers. They should be removed from the fire once or twice daily. It is not necessary to stir the fire so completely as to waste the coal through the grate. With a good chimney draft, some ashes just above the grate line will be a benefit in that it will retard the fire and tend toward less clinkering. *Clinkers are formed* with high volatile coals and strong draft through the grate. *They are avoided by slow and steady combustion, by having a thick fuel bed of live coals and by having slow draft through the grate (generally draft damper fully closed and small draft above the fire).* The arrangement of these dampers will be determined by experience.

A good sized chunk of wood embedded into the top of the fuel bed is a coal saver.

When replenishing a poor fire do not shake the fire, but put on some coal (or chunk of wood) and open the drafts. After the fuel is well ignited clean the fire.

The ash pit should be cleaned each day. An accumulation of ashes below the grate soon warps the grate and burns it out. Sifting shovels may be used and the unburned coal put back in the furnace.

Keep all dampers in working order.

Have a hand damper in the smoke pipe and keep it open only as far as is necessary to create a draft. Check damper (opening to basement air) must not be open unless draft damper under grate is closed.

Keep the water pans full of water and all humidity apparatus working.

Clean the base of the chimney, the furnace and the smoke pipe thoroughly in all parts at least once each year.

Keep the fresh air duct free from rubbish and impurities.

Allow plenty of pure fresh air to circulate through the furnace. In cold weather part of this supply may be cut off. When fuel saving is a necessity, it may be cut off entirely.

Have the basement well ventilated by means of outside wall ventilators, or by special ducts leading to the attic. Never permit the basement air to be circulated to the living rooms.

To bank the fires for the night, shake down and clean the fire, bank the live coals to one side of the fire box, fill up with fresh fuel, sift the ashes and distribute the unburned coal on the fire; with a poker make a hole through the fill into the live coal bed to permit of some flame above the fuel bed, close the under drafts and open the fire door draft slightly. Caution.—*Never cover the entire incandescent fuel bed with fresh coal and close the drafts.* If this is done, coal gas will collect above the fire and will ignite from the first flame that breaks through the fuel bed, causing an explosion.

CHAPTER VI.

HOT WATER AND STEAM HEATING.

DESCRIPTION AND CLASSIFICATION.

78. Hot Water and Steam Systems Compared with Furnace Systems:—Hot water and steam installations are more complicated in the number of parts than furnace installations; they use a more cumbersome heat carrying medium, for which a return path to the boiler must be provided; and have parts, in the form of radiators, which occupy valuable room space. But the hot water and steam plants have the advantage in that the circulation, and the transference of heat, are not affected by wind pressures. Hot water and steam will carry heat as readily to the windward side of a house as to the leeward side, a point which is known to be quite impossible with air. Furnace heating has the advantage of inherent ventilation, while the hot water and steam systems, as usually installed, provide no ventilation except that due to air leakage.

79. Elements of Hot Water and Steam Systems:—Hot water and steam systems consist of three principal parts: the boiler or heat generator, the radiators or heat distributors, and the connecting pipe lines which provide the circuit paths for the hot water or the steam. In the hot water system it is essential that the heat generator be located at the lowest point in the circuit for, as explained in Art. 11, the only motive force is that due to convection currents in the water. In the steam system this is not essential. The water of condensation may or may not be returned by gravity to the boiler. Hence, with a steam system a radiator may be placed below the boiler, if its condensation be trapped or otherwise taken care of.

Concerning piping systems and connections, several terms commonly used by heating engineers should be defined. The large pipes in the basement connected directly to the source of heat, and serving as feeders to the pipes running vertically in the building, are known as *mains*. *Supply mains* are those that carry water or steam from the source of heat to the radiators and *return mains* are those

that carry water or condensation from the radiators to the source of heat. The vertical pipes connecting between floors are called *risers,* while the short horizontal pipes between risers and radiators are *riser arms* or *branches.* As there are supply mains and return mains, so also there are supply risers and return risers. A return main traversing the basement above the water line of the boiler is designated a *dry return* and carries both steam and water of condensation; one in such position below the water line as to be filled with water is designated a *wet return.* The returns of all two-pipe radiators connecting with wet returns are said to be *sealed.*

80. Classifications:—*One classification* of hot water and steam systems is based upon the position and manner in which the radiators are used. The arrangement which is most familiar is the one wherein the radiators are located within the space to be heated and are surrounded only by room air. Radiators so placed (Fig. 40) provide no ventilation and are designated *direct radiation.* In *direct-indirect*

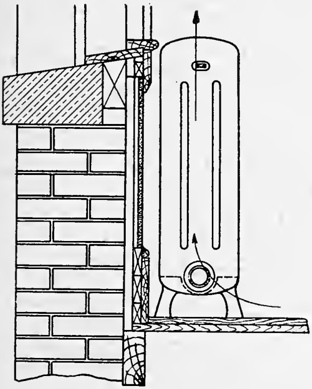


Fig. 40.

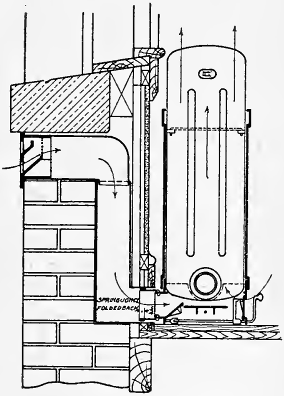


Fig. 41.

radiation the radiators are placed as in direct radiation but the lower portion of each radiator is encased and connected with the outside air as shown by Fig. 41. The direct-indirect system provides certain ventilating possibilities and should always be used in connection with inside wall ventilating stacks. *Indirect radiation* is installed remote from the rooms

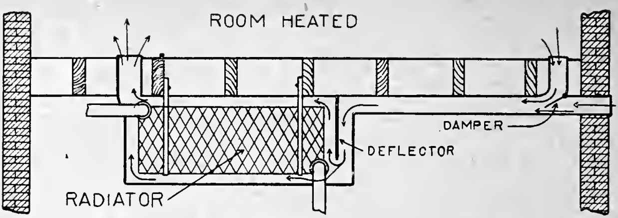


Fig. 42.

to be heated and ducts carry the heated air from the radiators to the rooms either by convection, or by fan or blower pressure. In residence work this radiation is usually suspended from the basement ceiling as shown by Fig. 42. This

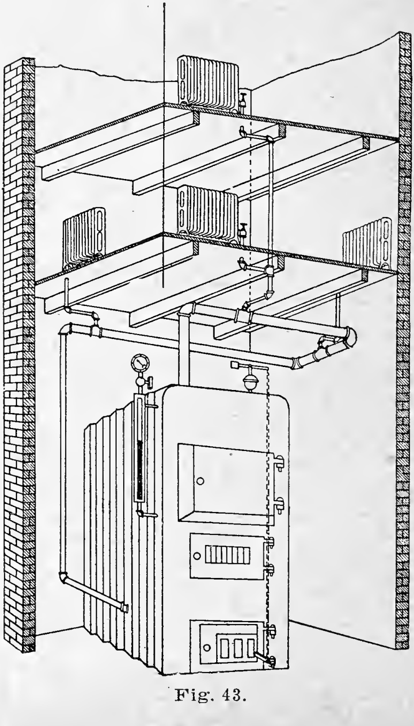


Fig. 43.

provides a combination system of steam and indirect warm air. When the radiation for an entire building is installed in one basement room, and each room of the building has carried to it its share of heat by forced air through ducts from one large centralized fan or blower, the system is called a *plenum system* or *fan-coil system* and is given special consideration in Chapters X to XII.

A second classification for hot water and steam systems is made according to the method of pipe connection between the heat generator and the radiation. The *one-pipe basement main steam system* (Fig. 43) is the simplest in construction and is preferred by many for steam installations. As the name indicates, its distinguishing feature is the single pipe path leading from the source of heat to the radiator, the steam and the returning condensation both using this path. In the risers and connections the steam and condensation flow in opposite directions, thus requiring larger pipes than where a flow and a return are both provided. In the mains the condensation usually flows with the steam and not against it. In the so-called *one-pipe basement main hot water system* (Fig. 49), radiators have two tappings and two risers, but the flow riser is tapped out of the top of the single basement main, while the return riser is tapped into the bottom of that same main by either of the special fittings shown in section in Fig. 44. The theory is that the hot water from the boiler travels along the top of the main, while the cooler water from the radiators travels along the bottom of this same main and two streams remain separate. Where mains are short and straight as in small residence installations, this system seems to give satisfaction, but where mains are long and more complicated a mixing of the two streams is unavoidable and the supply to the farther radiators is cooled to such a degree that the system becomes unreliable.

Fig. 44.

The *two-pipe basement main system* (Figs. 47 and 50) is standard with both steam and hot water installations. For steam work (especially for small installations) it is probably no better than the one-pipe system but for hot water work it is much preferred. In this system two separate and

distinct paths may be traced from any radiator to the source of heat. In the basement are two mains, the supply and the return, and the risers from these are always run in pairs, the supply riser on one side of a tier of radiators, the return riser on the other side. A two-pipe steam system should have sealed returns (See Art. 82).

The attic supply system, or Mills system, has found much favor with heating engineers in the installation of the larger steam and hot water plants. In this system the supply and returns both flow downward. This is accomplished by first leading the steam or water to the attic through one large main which there branches to supply the various risers. One riser only is generally used for each tier of steam radiators. Fig. 45 shows one- and two-pipe radiator connections. Frequently two-pipe connections are made to a single riser pipe. When this is done a water type radiator must be used with the supply entering the top and the return leaving the bottom of the same side (See vapor heating systems).

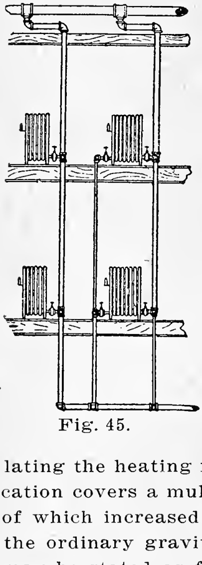


Fig. 45.

A third classification may be made, having reference to the manner of circulating the heating medium and to its pressure. This classification covers a multitude of inventions upon the attachment of which increased capacity and efficiency are claimed over the ordinary gravity systems. In outline, this classification may be stated as follows:

Gravity Systems

Steam systems, circulating steam at pressures greater than atmosphere.

Water, open tank systems, circulating water by increased weight of water in return risers over warm water in supply risers.

Modified Gravity Systems

Steam systems, circulating steam at atmospheric pressure or below.

Water systems, circulating water under pressure, at temperatures above those possible with the open tank systems and with accelerated velocities.

Combination steam and gas systems with radiators as heaters independent of a centralized heat supply.

Systems mentioned in this classification are explained in Arts. 81 to 85.

GRAVITY SYSTEMS.

81. Steam and Hot Water Systems:—Ordinary low pressure steam installations operate at pressures from 1 to 10 pounds gage. Relief valves are provided which release the steam when pressures tend to increase above the set maximum and thus protect the boiler from excessive pressures. Pressures in the boiler are maximum. These decrease gradually along the circuit of the supply and return mains because of the frictional retardation of the circulating steam, giving a pressure drop between main and return near the boiler of ½ to 1 pound. The water in the return, therefore, stands above the water level in the boiler an amount sufficient to balance this differential pressure. All pipes in the system are graded for easy flow of the condensation back to the boiler. Each boiler must be fitted with a pressure gage, a safety valve or pop valve and a draft regulating device. Each radiator must have a first-class automatic air valve.

An ordinary hot water installation has an *open expansion tank* at the highest point of the system to permit change in volume in the water as it changes temperature, such systems operate at pressures equivalent only to the static head of the water in the system. Pressures at the boiler range from 15 to 25 pounds gage for residence work. Water temperatures above 212°, therefore, will cause a loss of steam out the overflow of the expansion tank and are not considered advisable. Each boiler is fitted with a pressure gage or altitude gage to show the height of water in the expansion tank, a thermometer to show the temperature of the circulating water and a draft regulating device. Each radiator must have a compression air cock.

82. Diagrams for Gravity Steam and Hot Water Piping Systems:—Figs. 46 to 51 inclusive show some of the methods of connecting up piping systems between the source of heat and the radiators. A, B, C and D show different methods

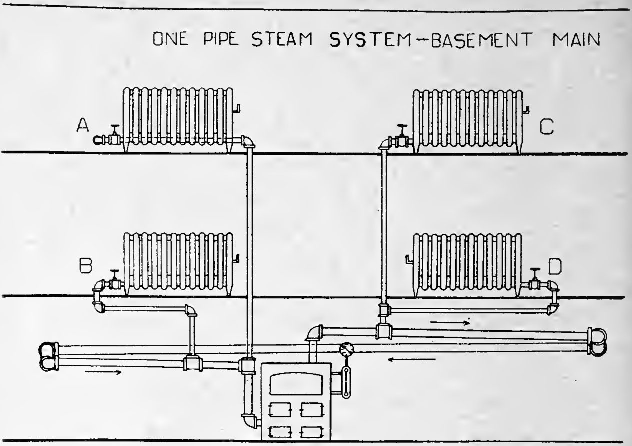

ONE PIPE STEAM SYSTEM—BASEMENT MAIN
A
C
B
D

Fig. 46.

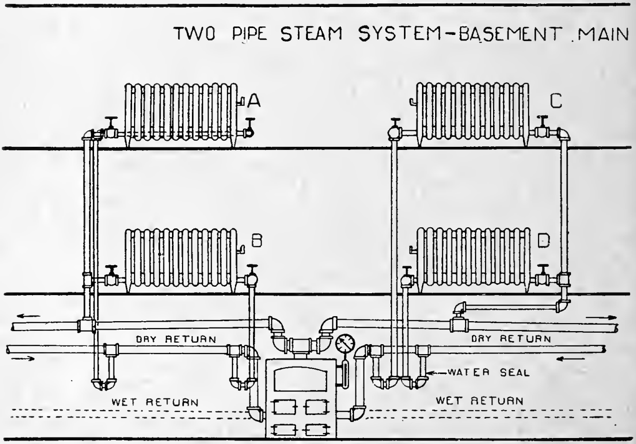

TWO PIPE STEAM SYSTEM—BASEMENT MAIN
A
C
B
D
DRY RETURN
DRY RETURN
WATER SEAL
WET RETURN
WET RETURN

Fig. 47.

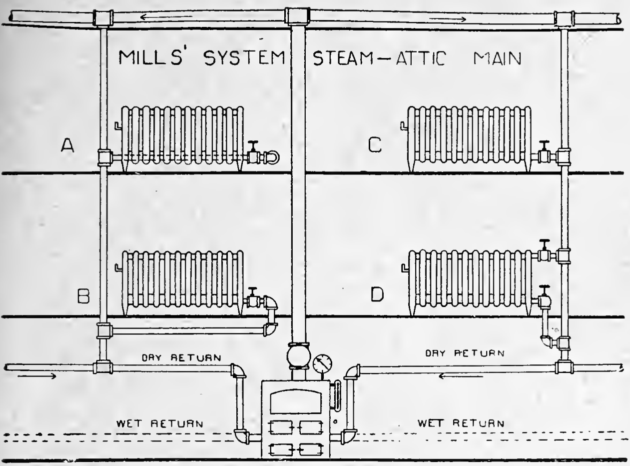

MILLS' SYSTEM
STEAM—ATTIC MAIN
A
B
C
D
DRY RETURN
DRY RETURN
WET RETURN
WET RETURN

Fig. 48.

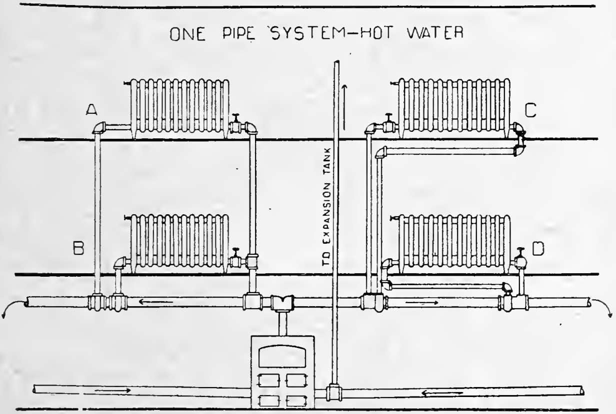

ONE PIPE SYSTEM—HOT WATER
A
B
C
D
TO EXPANSION TANK

Fig. 49.

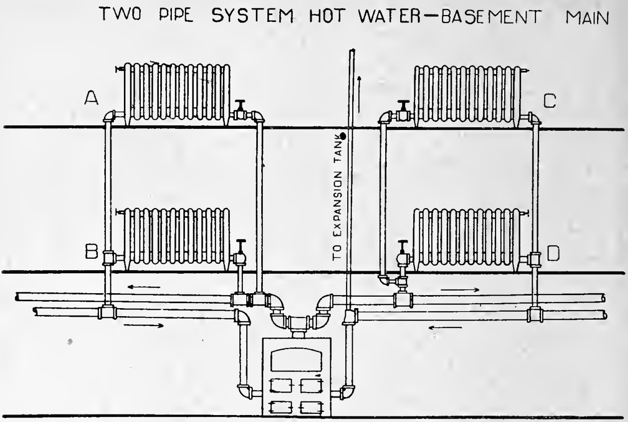


Fig. 50.

of connecting between the radiators and mains. The branches below the floor and behind the radiators are for the purpose of taking up expansion. Short connections should be avoided. It will be noticed that the two-pipe steam systems have sealed returns where they enter the main return above the water line of the boiler. *Dry returns* frequently interfere with the circulation of the steam to the radiators by *short-circuiting*. Steam from the boiler follows the path of least resistance to each radiator and many times this path leads up the return line into the radiator instead of through the supply. In Fig. 47 suppose the radiators C and D increased in number to the right C′, D′, C″, D″, etc., and all connected to the main as shown and to the return without the loop. It is easy to see that steam from the supply main would flow through the radiators C and D into the dry return where it would continue to the end of the line and affect the easy flow of steam through the end radiators. If the main inlet to any radiator were restricted, the steam to that radiator would be supplied through its return branch thus blocking circulation and causing *water-hammer*. The only way to insure against this is to *water-seal* each return by connecting as shown or by connecting to a wet return.

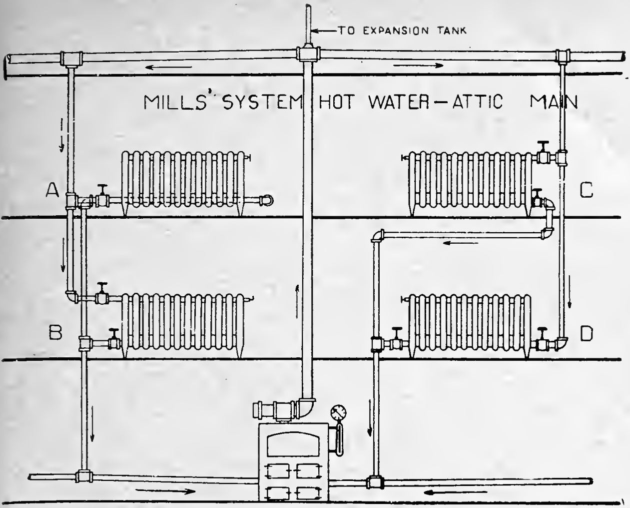


Fig. 51.

Hot water gravity circulation is more easily retarded than steam circulation and greater care must be exercised in laying out and installing the systems. Fig. 50 shows the connections most frequently used with basement mains. Connections recommended for the Mills overhead hot water system are shown in Fig. 51. Where all radiators in the same tier are connected flow and return to the same drop riser, circulation is frequently equalized in the radiators by O. S. Distributors turned against the stream in the supply and with the stream in the return (See Fig. 52).

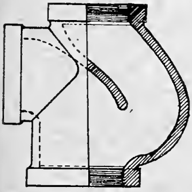


Fig. 52.

Basement radiation usually has poor circulation. In steam systems if the water of condensation is to be returned to the boiler it is placed on the ceiling or wall as high above the water line as possible. If the water is to be trapped to the sewer it may be placed on the floor. Hot water radiation may be placed at any elevation above (not below) the return inlet to the boiler. Circulation is improved, however, if the radiator supply is connected

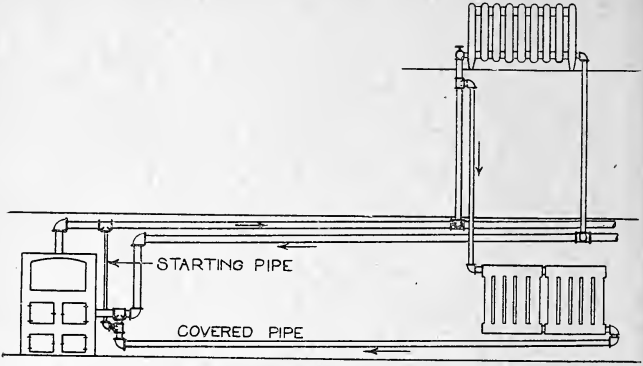


Fig. 53.

from some point above the basement. Fig. 53 shows such connections. Increasing the drop increases the rate of circulation.

MODIFIED GRAVITY SYSTEMS.

83. Atmospheric and Vapor Systems:—Low pressure steam systems are not as well adapted as hot water systems for moderate service, say on spring and fall days when only a small percentage of the full capacity of the heating system is required. Difficulty is experienced in keeping uniform temperature conditions in the radiators. In small plants, such as are found in residences where a constant attendant can not be provided, temperatures alternate rapidly between maximum and minimum. In an endeavor to meet the demand for a steam system which will serve for all outside weather conditions, a number of modified low-pressure steam systems, called vacuum, vacuo-vapor, vapor, modulation or atmospheric systems have been devised. It is claimed for these systems that they give better regulation and more uniform temperature conditions, also that they are free from air troubles.

The term *vacuum* should properly not be applied to this class but should belong to those systems having a positive vacuum in the returns mechanically produced by action of

pumps, ejectors, etc., as explained in Chapter IX. There is one gravity system, however, that has some claim to the name—the *Mercury Seal Vacuum System.* Fig. 54 represents the outer coils of any radiator. Inside the last coil is a mercury pot with a vertical iron tube connection for mercury column similar to the average barometer. The top of this tube is connected with the atmospheric side of the automatic air valve. When not in service the mercury in the column drops to the pot. When firing up, the air valve permits the escape of air but closes against steam. The mercury pot freely allows the escape of this air but does not permit its return. As a result the heating system (any kind of system) warms up and expels the air but when it cools down a partial vacuum is established and the water continues to boil at temperatures below 212°. If the system of piping and valves is very tight a partial vacuum may be maintained throughout the night, during which time steam will circulate at low pressure until the temperature falls say as low as 150°. Further it will heat up more quickly and with less fuel in the morning because of this partial vacuum. To have a mercury column at each radiator would be prohibitive because of the expense, consequently where this system is used the air lines are run to the basement in a similar way to those of the returns, collected together and attached to a mercury seal of sufficient size to expel the air from the entire heating system. Such a system is sometimes called an *air-line system.* The above arrangement is very desirable and is in sharp contrast to the ordinary system where the steam leaves the radiators as soon as the temperature falls below 212°. The practical difficulties in obtaining and maintaining an air tight piping system, however, limits its use.

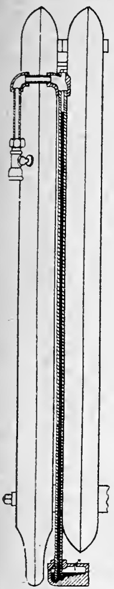


Fig. 54.

The terms *vapor, vacuo-vapor, modulation, atmospheric* and the like are trade terms that are not especially distinctive but which indicate all the large number of gravity steam systems operating at pressures from 0— to 1— pound gage. In these systems the radiators are water-type, two-pipe,

top connected and have packless valves and no air valves. Steam being lighter than air it first fills the top of the radiators and gradually forces the air downward and out the return to the atmosphere. The radiator outlet is usually on the opposite side of the radiator from the inlet, water of condensation and air both passing through this opening.

The important feature in operating any gravity heating system at pressures near atmosphere and especially those having no automatic control on the air relief, is *effective regulation.* Where this is obtained there will be fairly uniform temperature conditions within the radiator, sufficient heat emission to satisfy the room requirements, and no wastage of steam. Regulation may be applied at any of the four points in the system—at the boiler, in which case the drafts are controlled by a hydraulic head, a float located in a receiver at the end of the return main or by a pressure regulator connected to the steam space of the boiler—at the inlet valve to the radiator—at the radiator outlet—and at the atmospheric vent at the end of the return main. All systems of this kind have automatic draft regulation and all have radiator inlet valves that give more or less satisfactory hand or thermostatic control. The essential differences in the various types, therefore, lie in the character of the regulation at the radiator outlets and at the air relief on the end of the return main. Classifying the many systems on the market, a few only of which will be mentioned, they may be grouped under three general heads.

Type 1, Fig. 55, has no positive regulation on the radiator outlets or on the air relief. (A thin water seal is here considered as no regulation since in every case a positive vent opening is provided for air). In estimating radiation for this type, 20 per cent. more is put in than would be required for any low pressure steam system with closed returns. This extra radiation serves to condense the steam that may be admitted to the normal radiator beyond its required condensing capacity. If too much steam is admitted for any given outside temperature it will pass into the returns and out into the air. *In this system, therefore, it is very desirable that the best of regulation be applied to the draft dampers at the boiler* and also that careful adjustment be made on the valve inlets to the radiators. Since most vapor inlet valves are hand operated and are subject to the eccentricities of the attendant, too much dependence should not be

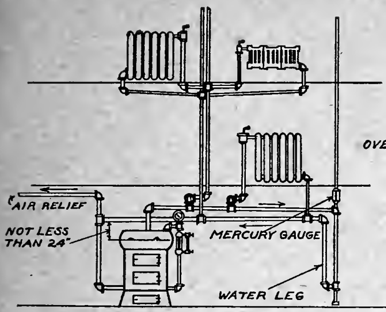


Fig. 55.

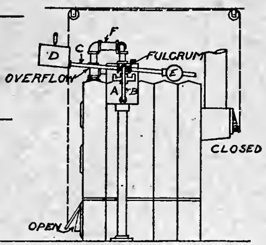


Fig. 56.

placed in this regulation. It will be noticed that the end of the main is separately vented and enters the dry return through a water seal. This serves to cut off direct steam circulation into the return. Two representative systems of this class are the *Atmospheric* and the *Mouat-Squires*.

The damper regulator of the Mouat-Squires system is worthy of special mention. Fig. 56, tank *A* is filled with water to the overflowing point, the overflow connection being opposite the fulcrum. Steam enters the regulator from the boiler through connection *F*, forces the water down in tank *A* and up through the flexible hose *B* and the hollow lever *C* into tank *D*, causing tank *D* to drop when a sufficient weight of water to overcome the weight of the counterbalance *E* has entered same. This causes the drafts to close. When the pressure decreases in the boiler, the water returns to tank *A* by gravity, causing the reverse operation of the regulator and dampers. The setting of the counterweight *E* regulates the vapor pressure at which action takes place.

A slightly modified form of Type 1 (Broomell System Fig. 57) has a receiver at the end of the return main at the boiler and an air relief from the top of the receiver to the atmosphere. The air relief leads through a condenser to condense and return to the boiler any steam leaving with the air. The end of the main may be separately vented or connected with the air relief. Fig. 58 shows two sections of the receiver. A copper float rides on the water in the re-

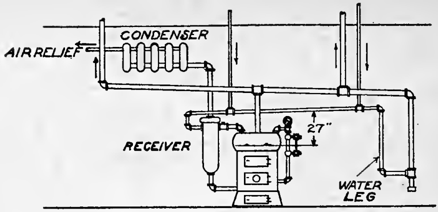


Fig. 57.

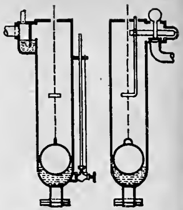


Fig. 58.

ceiver and is connected by chain to the dampers. The level of the water in the receiver remains the same as that in the boiler as long as all the steam generated in the boiler is used in heating. When excess steam is generated the pressure increases, the water level rises in the receiver and the float closes the drafts. If the float rises high enough to lift the adjusting rod, it unseats a safety valve and blows off the steam. The reverse action takes place when the pressure drops.

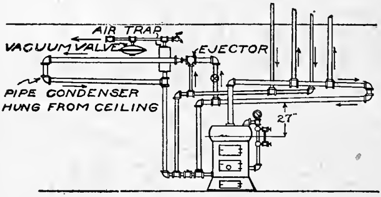


Fig. 59.

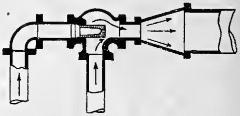


Fig. 60.

Type 2, Fig. 59, has no regulation on the radiator outlets (radiators not shown) but has a condenser coil and thermostatic control on the air relief which connects with both main and return. By the use of a check valve or mercury seal beyond the thermostatic valve a partial vacuum may be temporarily produced in the system. In this type the amount of radiation is normal and the steam pressure may rise above normal with automatic air release without waste of steam. The *Moline System* is typical of this class. Note the ejector, Fig. 60. This is supplied with steam from the end of the supply main and ejects the air and vapor

from the end of the return main into the condenser, from which the air is released through the air trap and the condensation is returned to the boiler.

Type 3, Fig. 61, has a normal amount of radiation, a positive thermostatic control on the radiator outlets and an air relief connecting with the ends of the main and return either mechanically or thermostatically controlled. Three representative systems of this class are the *Dunham, Webster* and *Illinois.* Attention is called to the equalizer pipe be-

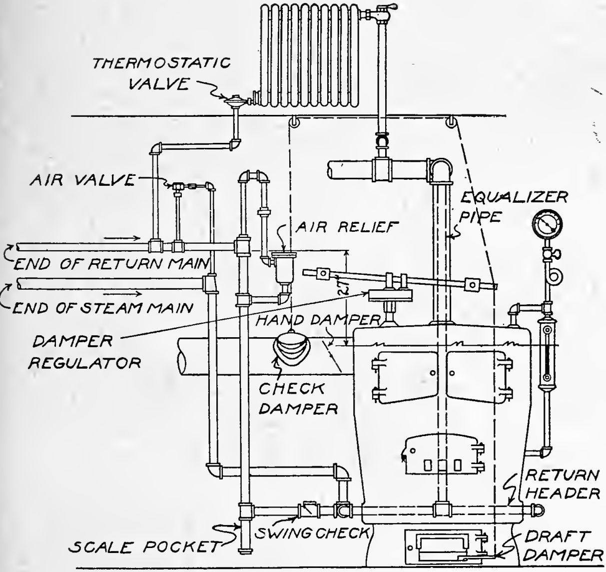


Fig. 61.

tween the main and return at the boiler, the swing check between the return riser and the boiler and the scale pocket on the bottom of the return riser to protect the check from scale and dirt. Also notice the difference in levels between the normal water line in the boiler and the lowest end of the return main. This requires a water column of at least 27 inches to provide sufficient head to overcome the inertia of the check and to account for a small differential pressure

between main and return. The air relief in this type as in the type preceding may be made to close under pressure and pressures above normal may be used. Type 3 gives a more positive circulation in the mains and radiators, less danger of short circuits and greater pressure range than those systems not equipped with thermostatic valves on radiator outlets. Steam pressure regulators, similar in action to Fig. 62 are used for damper control.

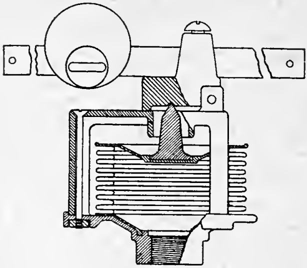


Fig. 62.

Atmospheric and vapor heating systems that may at times be operating under pressures varying from 2 to 10 pounds gage are fitted with *return traps* that close the air relief when the differential pressure between main and return reaches a fixed amount. At the same time live steam is automatically admitted to the top of the trap forcing the collected return water through the check into the boiler. The air vent remains closed and action continues as an ordinary closed steam system until the differential pressure falls to normal when the action is reversed and it again becomes an open relief atmospheric system. The Webster return trap (Fig. 63) shows one of the simplest forms of these traps.

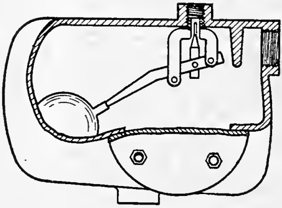


Fig. 63.

84. Modified Open Tank Hot Water Systems:—A number of modifications have been adapted to low pressure hot water heating systems for the purpose of increasing the temperatures, pressures and velocities of the circulating water above those obtained by the open tank system. Out of a large number of systems four of these will be mentioned as type representatives. Increasing temperatures permits a reduction in radiation so as to compare with that of steam systems. This is desirable since large radiators are an obstruction in any room. With increased velocities pipe and fitting sizes may be reduced. This also is very desirable in any

system from the standpoint of adaptability. In addition any reduction of this kind causes a reduction in first cost.

In the *Honeywell System* (Fig. 64) a purely American system, a mercury seal tube is connected between the upper point of the main riser and the expansion tank. This is designed to hold a pressure within the system at that point of about 10 pounds gage. Water from the system fills the casement and presses down upon the top of the mercury in the bowl. Increasing the pressure in the system lowers the level of the mercury in the bowl and forces the mercury up the central tube *A* until the differential pressure is neutralized by the static head of the mercury. If the pressure becomes great enough to drop the level of the mercury to the tube entrance, water and steam will force through the mercury to chamber *D* and from thence through the expansion tank to the over-flow. Any mercury forced out of tube *A* by the velocity of the water and steam, strikes deflecting plate *C* and drops back through annular opening *B* to the mercury bulb below. As the pressure is reduced in the system the mercury drops in tube *A* to the level of that in the bulb and water from the expansion tank passes down through the mercury seal into the heating system to replace any that has been forced out of the expansion tank. This action is automatic and is controlled entirely by the pressure within the system. The only loss, if any, is that amount of water which goes out the over-flow. A similar arrangement is used in the *Cripps System*. In this the mercury seal is placed beyond the expansion tank and puts the expansion tank under pressure.

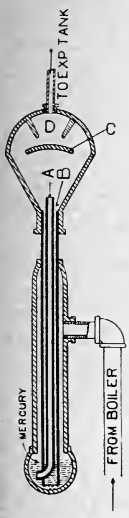


Fig. 64.

The extra pressure made possible by the Honeywell or Cripps apparatus makes it possible to carry the circulating water at temperatures as high as 240°, which is above that of the average low pressure steam system. With temperatures as high as this there is undoubtedly an increased differential temperature between flow and return which would tend to increase the velocity of the water and make it possible to reduce pipe sizes.

The *Koerting System* (Fig. 65), invented by German engineers, is an open tank system with a series of motor pipes leading from the upper part of the heater to a mixer, where the steam which has been formed in the heater and motor pipes is condensed by part of the circulating water entering through the by-pass from the return. The velocity of the steam and water through the motor pipes and the partial vacuum caused by the condensation in the mixer produces an acceleration up the flow pipe.

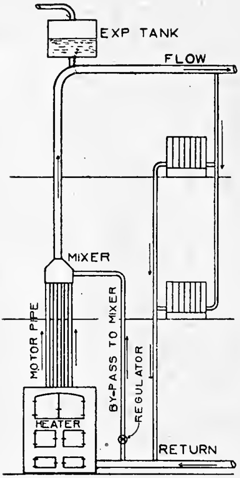


Fig. 65.

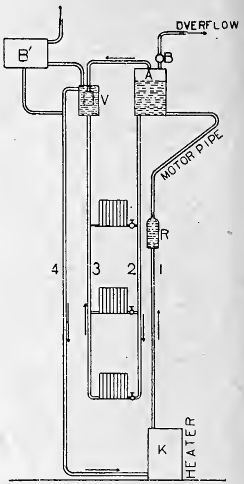


Fig. 66.

The *Bruckner System* (Fig. 66), invented by an Austrian engineer, is an open tank system with two expansion tanks. Heater *K* delivers the hot water (above 212°) up the flow pipe to receiver *R*, where a separation takes place between the steam particles and the water, thus causing an acceleration up the motor pipe to expansion tank *A*. The water in flow pipe *2* has a temperature slightly below that in *1*. After passing through the radiators the water in *3* is at a lower temperature than that in *2*. The steam particles which have collected in expansion tank *A* above the water line are condensed in *V*. The acceleration in the system is thus produced by a combination of the upward movement of the

steam particles in motor pipe *1* and the induced upward current in *3* toward condenser *V*. It will be noticed by comparing with Fig. 65 that the condensation in one system takes place before the expansion tank and in the other system after it has passed the expansion tank. Each of the systems illustrated may be carried under pressure by applying a safety valve as at *B*, a mercury column as in Fig. 64, or by an expansion tank located high enough to give sufficient static head.

The *Reck System*, invented by a Danish engineer, is illustrated by Figs. 67 and 68. Water passes from the heater

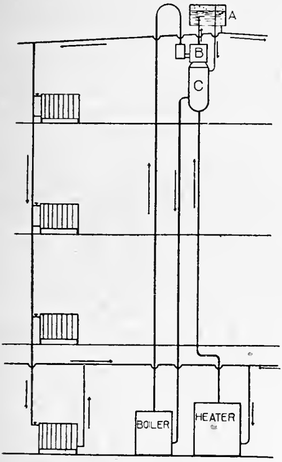


Fig. 67.

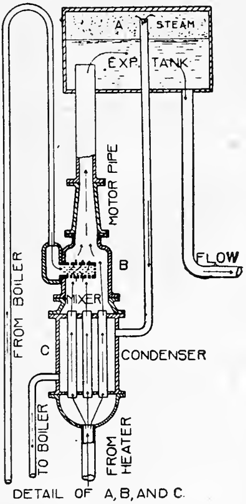


Fig. 68.

up the main riser to condenser *C* and thence into expansion tank *A* as a supply to the flow pipes of the system. Steam from a separate boiler is admitted to mixer *B* above the condenser and enters the circulating water just below the expansion tank. The velocity of the steam and the partial vacuum caused by the condensation induces a current up the flow pipe to the expansion tank. When the water level in the expansion tank reaches the top of the overflow pipe the water returns to the steam boiler through condenser *C* where

it gives off heat to the upper current of the circulating water. It will be seen that the circulating water in the system and the steam from the boiler unite from the inlet at the mixer to the expansion tank. On all other parts of the systems they are independent.

Fig. 69 is a modification of this same principle, wherein air is injected in the riser pipe at *B* and causes acceleration by a combination of the partial vacuum produced by the steam condensation as just mentioned and the upward current of the air particles as in an air lift. Steam enters through pipe *J* and ejector *H* to the mixer at *B* where it is condensed. In passing through *H* air is drawn from tank *E* and enters the main riser with the steam. The upward movement of this air through the motor pipe to the tank induces an upward flow of the water in the main riser. By this combination there are formed three complete circuits, water, steam and air, uniting as one circuit from the mixer *B* to expansion tank *E*. The steam furnished in principle 3 may be supplied by a separate steam boiler or by steam coils in the fire box of a hot water boiler.

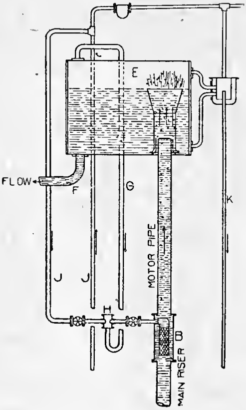


Fig. 69.

Acceleration is also produced by some piece of mechanism as a pump or motor placed directly in the circuit. This principle is discussed under District Heating and will be omitted here.

85. Gas- and Electric-Steam Heating Systems:—A *gas-steam* system of heating, similar in many respects to the Rector gas radiator system (Art. 74), is frequently used. In this case the heating unit is a combination gas stove and steam radiator. The gas supply (either natural or artificial) is automatically controlled by a diaphragm valve from the pressure within the radiator. Radiators without exhaust pipes may be used with artificial gas for limited heating, but they should be supplied with exhaust pipes in every case where natural gas is used and where artificial gas is used in amounts to render the room air impure. An *electric-steam*

system is sometimes used for the same service. The radiators are made of pressed steel, electrically welded and sealed. The radiator contains a small amount of distilled water (a 6-section type having about a quart which never needs replacing). The electric heating unit is about the same capacity as those used in electric flat irons. The heating unit, partially surrounded by water and under partial vacuum, heats readily. Systems of this type are in use in climates where only moderate heating is necessary.

86. Piping Connections:—Many heating systems have

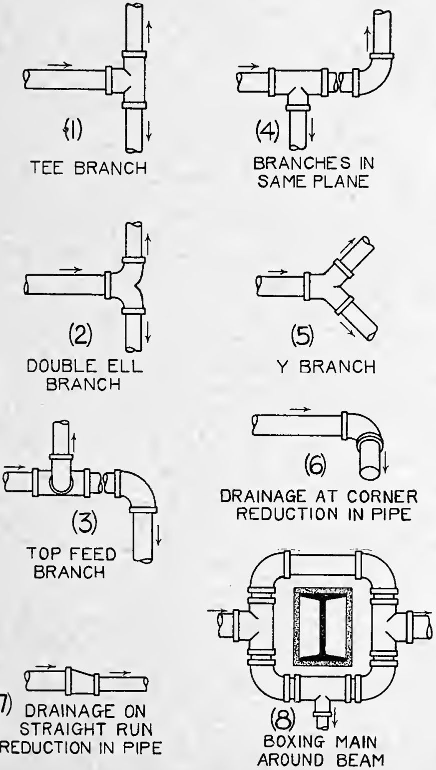


Fig. 70.

been crippled by improper piping connections. Figs. 70 and 71 show some of the standard forms. In this connection a few suggestions may be valuable. (1) A steam main may branch right and left through a straight tee providing the lineal expansion of the branches is provided for. (2) Right

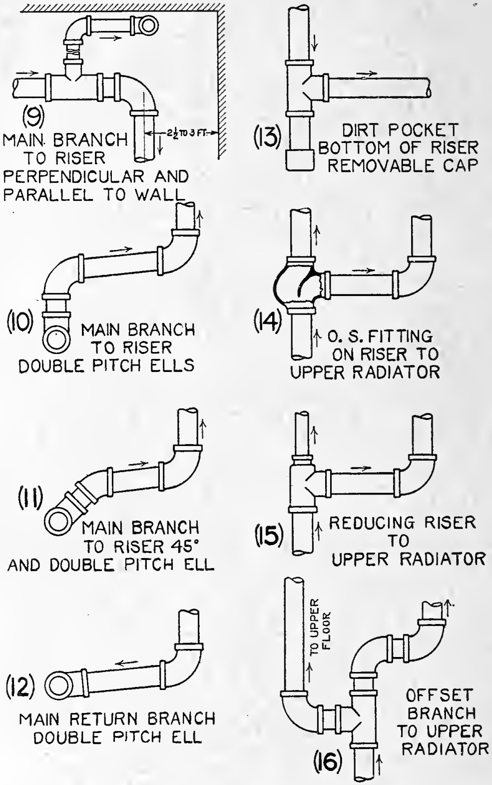


Fig. 71.

and left branches through straight tees in hot water systems should never be used. A double sweep ell should be used instead, this will divide the two streams of water without causing eddy currents. (3) Any branch that is to be favored should be taken from the top of the main by vertical

or 45° lines. No hot water branches should ever be taken off the side of the main. (4) Offset branches provide expansion facilities. (5) Hot water mains may branch through Y fittings. This is ideal for circulation but does not absorb expansion as readily as 90° turns. (6, 7) Steam mains which change size should be graded on the bottom for satisfactory drainage. This may be done at a corner by a reducing ell pitched slightly downward or on a straight run by an eccentric fitting. (8) Steam mains may pass an obstruction by boxing around the obstruction, the drop providing drainage and the rise the steam circuit. (9) Mains are kept 2½ to 3 feet from the wall, well supported from the ceiling and free to move in any direction to allow for expansion. The corners of the main should not be anchored by running diagonally to the riser. Branches should be run perpendicular to and parallel with the wall. (10, 11, 12) Double-pitch ells should lead to risers in all places where necessary. Long branches to risers are to be avoided where possible. (13) Dirt pockets should be provided at the bottom of return risers where there is danger of clogging valves. (14, 15, 16) Radiation on the upper floors will rob the lower floor radiation, consequently retarding influences such as O. S. fittings, reducers and offsets should be put in to give advantage to lower floors. (17) Radiators should be connected with branches sufficiently long to take up expansion. (18) Water pockets should be avoided in horizontal mains and branches. All pipes should be well pitched for drainage. (19) Risers may be run within the wall or in closed chases in the face of the wall for appearances. Complete encasement within the wall, however, should be made only with the knowledge and consent of the owner since in many cases walls have been ruined by defective pipes. (20) Branches may also be run within the floor construction, but extra care should be used in the laying.

CHAPTER VII.

HOT WATER AND STEAM HEATING

BOILERS, RADIATORS, FITTINGS AND APPLIANCES.

87. **Steam Boilers and Water Heaters:**—Heaters for supplying hot water and boilers for supplying steam to heating systems may be divided into three classes: round vertical, having capacities of 250 to 1500 sq. ft.; sectional, having capacities of 300 to 9000 sq. ft.; and water tube or fire tube, having capacities of 10000 to 40000 square feet of direct steam radiation. Round and Sectional boilers (Figs. 72 and 73) are made of cast iron, are of the portable type and need no special casings other than the plastic coverings to reduce radiation. Fire tube and water tube boilers (Figs. 74 and 75) are of wrought iron or steel and are encased in brick work. Boilers of the largest capacities are water tube type and are always used in central station work. Heating boilers for residence work are usually of the sectional type. These boilers are very flexible and may be increased in capacity by adding sections to existing boilers to meet increased requirements. In some installations it is better to install two boilers of somewhat reduced capacity (say ⅔ of the calculated capacity) and either boiler will more nearly meet the average load. This is frequently done where break downs may cause serious inconvenience. In general it may be said that products of the various manufacturers show but little difference in design between hot water heaters and steam boilers and as a result the two types are usually referred to as *boilers.*

Boiler capacity depends principally upon the amount and arrangement of the grate and heating surfaces. Grate surface is the gross area of the fuel bed at the top of the grate. Heating surface refers to those boiler plates that have the fire or heated gases on one side and water on the other. Heating surfaces are of two kinds, direct and indirect (sometimes called prime and secondary). *Direct surfaces* are those so located as to receive the direct heat or radiant ray of

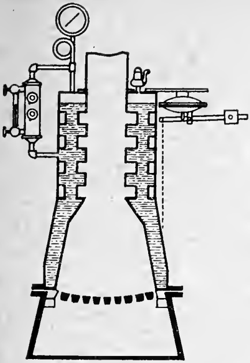


FIG. 72

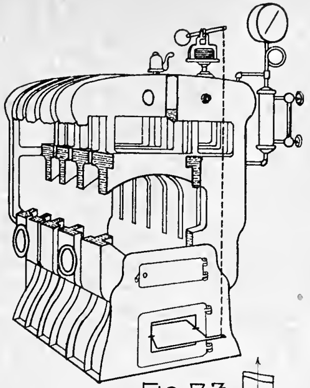


FIG. 73

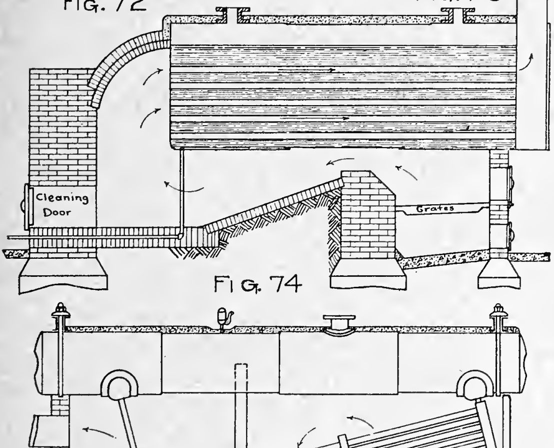


FIG. 74

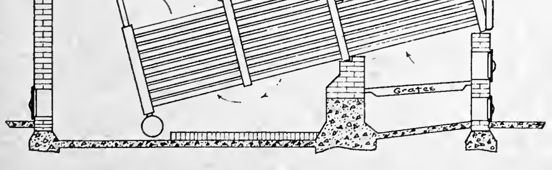


FIG. 75

the fire. Indirect surfaces are all those not included under direct, i. e., those that are in contact only with the heated gases of combustion. Direct surfaces transmit more heat per unit of time than the same area of indirect surface because of the greater difference in temperature between the two sides of the plate. For this reason boilers have as much direct surface as is possible to give them. The *average amount of heat transmitted* through boiler plates will vary from 1600 to 2500 B. t. u., for heating boilers and 2000 to 3000 B. t. u., for power boilers. The *rate of heat transmission* for clean metal surfaces should be practically the same for either direct or indirect locations. See also Art. 61 on furnace heating surfaces.

Proper combustion of the fuel and the most efficient transmission of the heat of the fire across the plates to the water are of prime importance. It is easy to see therefore

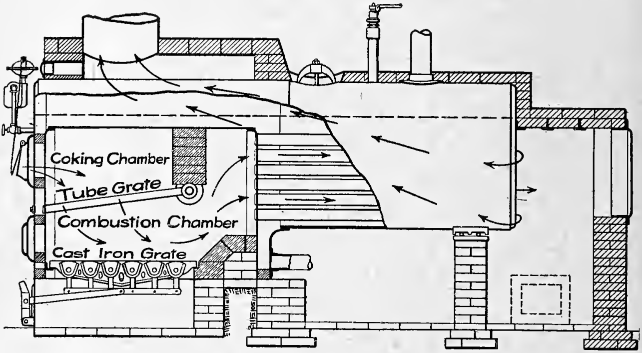


Fig. 76.

that the one feature of boiler design under continual study is the construction of the fire box or furnace. This is especially true of the boilers burning bituminous or soft coals. The average hand fired furnace, cared for by the average fireman is a nuisance in any business or residence district because of the smoke. In large plants the mechanical stoker which fires slowly and continuously at the front of the fire has proved the best remedy but in small plants where hand firing is necessary and where the fire is charged three to four times each twenty-four hours the problem is more difficult. A number of smokeless furnaces, represented by Fig. 76, have been developed along the lines of the

Hawley down draft furnace with water tube grates above the fire and fire grates below. Fuel is fed into the space *above* the water tubes (coking chamber) and there loses the volatile matter and hydrocarbon gases. These gases are practically all consumed in passing over the lower fire and through the length of the combustion chamber. The lower grate catches the coke product from the upper grate and is occasionally replenished by a charge of fresh coal near the front of the fire. These furnaces produce practically smokeless combustion and are being increasingly used.

The latest design of soft coal furnace is that shown in Fig. 77. This is of the sectional down-draft, grateless type and

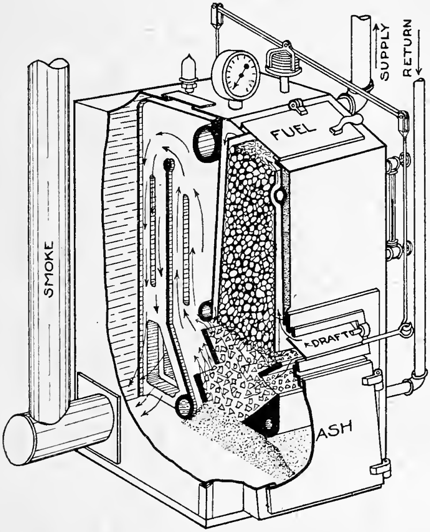


Fig. 77.

is especially designed for bituminous and soft coals, having a magazine above the fire which serves the purpose of supply box and coking chamber. In this arrangement combustion takes place as in the Hawley furnace, the liberated hydrocarbons being consumed while passing through the entire length of the combustion chamber to the chimney.

Round and sectional types of boilers have ratios of grate surface to heating surface varying between 1 to 15 and 1 to 25, and water tube or fire tube boilers varying between 1 to 40 and 1 to 60. The arrangement of the heating surface differs very much, each manufactured product having a distinctive design. According to Prof. Kent "for commercial and constructive reasons, it is not convenient to establish a fixed ratio of heating surface to grate surface for all sizes of boilers. The grate surface is limited by the available area in which it may be placed, but on a given grate more heating surface may be piled in one form of boiler than in another, and in boilers of one general form one boiler may be built higher than another, thus obtaining a greater amount of heating surface. The rate of burning coal and the ratio of heating to grate surface both being variable, the coal burning rate and the ratio may be so related to each other as to establish a rate of evaporation of 2 lbs. of water from and at 212° per sq. ft. of heating surface per hour."

Boilers may be selected by grate surface, heating surface, coal burned per hour, pounds of steam evaporated per hour and heating capacity in square feet of radiation (including mains). Manufacturers' catalogs give boiler ratings in terms of radiation supplied, with grate surface, heating surface and installation sizes for units of different capacities. The best method of selecting a heating boiler is to estimate the required grate surface of the boiler that will theoretically supply the given radiation and check this amount with the catalog data (See Art. 100). Considerable care must be exercised in the selection of the type of boiler to fit any given set of conditions. To illustrate: the grate and fire box should be designed favorable to the burning of the kind of coal that would be generally used; the boiler selected should permit of easy cleaning especially if it is a soft coal burner; the arrangement of the heating surfaces should be such that there will not be an excessive friction as the gases pass through the boiler; with an inside chimney there is little danger of lack of draft and any form of down draft boiler may be used, while with an outside chimney of ordinary construction there may be a question as to the use of such boilers; also, the kind of attention and the frequency of firing must be taken into account. For further study of boiler types and operations see Marks' M. E.

Handbook, Kent's M. E. Pocket-Book, Gebhardt's Steam Power Plant Engineering, Hirshfield and Barnhard's Elements of Heat Power Engineering, and trade catalogs.

Combination heaters are frequently installed to supply both warm air and steam or warm air and warm water to the same plant. For such systems a combination heater as shown in Fig. 26, Art. 65, is needed. It consists essentially of a warm air furnace with a steam or water radiator in the upper part of the fire pot. The radiator through the connected piping supplies heat to those sections of the building where satisfactory air circulation could not be had. The principal difficulty encountered in these combined systems is in obtaining the proper proportion of the heating surface of the furnace to that of the radiator to suit varying demands upon the system.

88. Boiler Accessories:—Water heaters are equipped with pressure gages or mercury columns for registering the pressures carried within the system, thermometers on the supply and return mains to give the differential tempera-

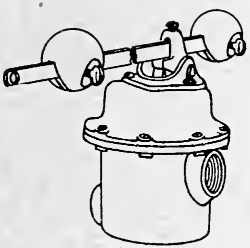


Fig. 78.

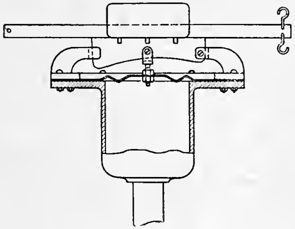


Fig. 79.

ture of the circulating water, and automatic draft apparatus controlled by thermo-regulation from the temperatures of the supply water or by a thermostat from the temperature of the room air. Steam boilers are supplied with pressure gages as in water heaters, safety valves or pop valves to relieve any excessive pressures, water glass and gage cocks to register the water levels, and automatic draft apparatus controlled by a diaphragm valve from the pressure of the steam in the supply main, by a float from the water level in the return main or by a thermostat from the temperature of the room air. Fig. 78 is a thermo-regulator for water systems. It operates from the elongation and contraction of a sylphon bellows enclosed within a cast iron casing.

The bellows, a brass, accordion pleated cylinder, is closed at both ends and contains a volatile fluid which vaporizes at low temperatures and causes varying pressure within the bellows. Water from the boiler circulates between the bellows and the casement and as the temperature of the water changes the state of the volatile liquid its pressure changes and the bellows increases or decreases in length and operates the draft. A modification of this type of regulator is used on steam system. In this the regulation is by steam pressure from the inside of the sylphon bellows (see Fig. 62).

Fig. 79 shows a diaphragm regulator which is usually attached to the steam space of the boiler or to the steam main close to the boiler. For details of specialties including glass gages, gage cocks, etc., see American Radiator and United States Radiator company's catalogs. For care of boilers and furnishings see Art. 115.

89. Radiators, Classification as to Material:—Radiators may be classified according to the materials used in their production as *cast iron, pressed steel* and *pipe coil.* Wall thicknesses of cast radiators are ¼ to $\frac{5}{16}$ inch. Pressed radiators are formed from sheet steel plates. Each section is composed of two pressed sheets that are welded together by a double seam around the edge and riveted between the columns. The sections of cast radiators are connected by mild steel or malleable *push* or *screw nipples* which serve as passageways between the sections for the heating medium. The malleable nipple is subject to occasional hidden defects from the process of casting but is not subject to corrosion as is true of the steel nipple, hence it is usually preferred. Pressed steel sections are welded together. Cast iron radiators have the disadvantage of weight and bulk and have a comparatively large internal volume, averaging a pint and a half per square foot of surface, but they are practically free from corrosion. Each radiator after being assembled is tested to 100 lbs. per sq. in. gage pressure. Pressed radiators have an internal volume approximating one pint per square foot of surface.

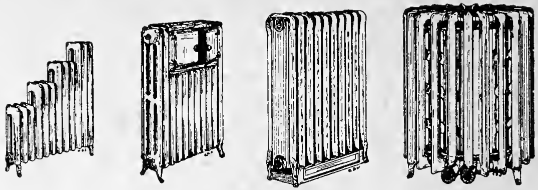
Radiators composed of pipes in various forms (vertical or horizontal) are commonly referred to as *coils.* They are not much used for direct or direct-indirect work because of the unsightliness. They are frequently used in indirect and plenum systems and are generally used in the direct heating of shops, factories and greenhouses. In coil heaters 1-inch

pipe is the standard size, however, in some cases (greenhouses) coils are used as large as 2 inches in diameter. Standard 1-inch pipe is rated at one square foot of heating surface per three lineal feet and has about one pint of containing capacity per square foot of heating surface.

90. Classification as to Form:—Radiators may be classified according to form as one, two, three and four column *floor types, wall type* and *flue type.* These terms refer only to cast and pressed radiators. The *column* of a radiator is one of the unit fluid-containing elements of which a *section* is composed. When a section has only one vertical unit it is called a single column or one column radiator, when it has more than one it is a two column, three column or four column type. End sections are called leg sections, intermediate ones are *loop sections*. The legs on all pressed steel radiators are detachable. A wall radiator is a one-column type, so designed as to be of the least practicable thickness. It frequently presents the appearance of a heavy grating and is designed to have 5, 7 or 9 square feet of surface, according to the size of the section. One column floor radiators made without feet are often used as wall radiators. A flue radiator is a very broad type of the one column radiator, the parts being so designed that the air entering between the sections from below is compelled to travel to the top of the sections before leaving the radiator. This type is well adapted to direct-indirect work.

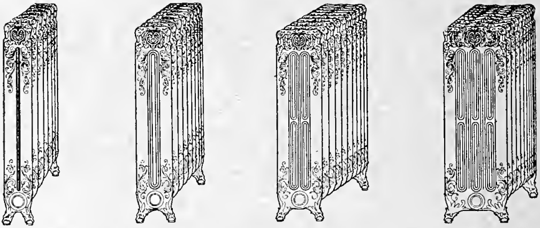
There are many special shapes of assembled radiators such as *stairway radiators* built up of successive heights of sections to fit along the triangular shaped wall space under stairways, *pantry radiators* built up of sections to form a tier of heated shelves, *dining room radiators* with an oven-like arrangement built in between sections, and *window radiators* built with low sections in the middle and higher ones at either end to fit neatly around a low window. Fig. 80 shows a number of these common forms used in practice. Figs. 81, 135-137, show methods of building up pipe coil heaters.

91. Classification as to Heating Medium:—A third classification according to the heating medium employed, gives rise to the terms *steam radiator* and *hot water radiator.* Casually one would notice little difference between the two, but in construction there is a vital difference. A steam radiator has its sections joined by nipples along the bottom only, but a hot water radiator has both top and bottom connec-



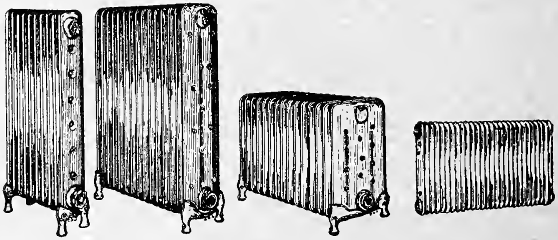
Stairway Type Dining Room Type Flue Type Circular Type

CAST RADIATORS



Wall Type Two-Column Type Three-Column Type Four-Column Type

PRESSED RADIATORS



Single-Column Type Two-Column Type Three-Column Type Wall Type

Fig. 80.

tions. This is quite essential to the proper circulation of the water. Steam type radiators are always tapped for pipe connections at the bottom. Hot water radiators may have the supply connections at the top and the return connections at the bottom, or both connections at the bottom. Hot water radiators can be heated very successfully with steam, but steam radiators cannot be used for hot water. Vapor systems are supplied with two-pipe, hot water type radiators.

Radiators *must have at least two tappings,* one below for the entry and exit of the heating medium, and one on the end section opposite (near mid-height for steam and at the top for hot water) for air discharge as shown by Figs. 46 and 48. They *may have* three tappings, a *supply,* a *return* and an *air* tapping as shown in Figs. 47, 49, 50 and 51 (For fittings see Figs. 88-92).

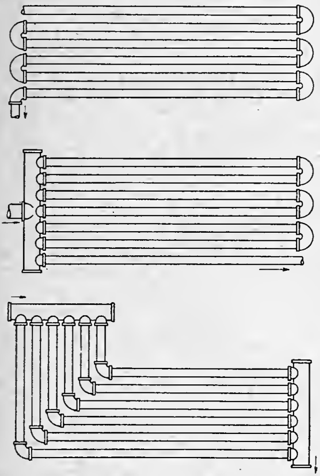


Fig. 81.

92. Effect of Height and Width of Radiator Upon the Transmission of Heat:—In selecting a radiator height for a given place the governing feature is usually the floor space allowed for the radiator. Thus, if a radiator 26 inches high requires so many sections that it is too long for the space allowed, a 32-inch or a 38-inch section may have to be taken. High radiators are less efficient than low radiators because as the air is heated in passing up the outside of the sections the differential temperature between the inside and the outside becomes less, and less heat is transmitted per unit area. By the same reasoning, horizontal pipes (coils) are more efficient than any other form of heating surface. Also, wide radiators are less efficient than narrow radiators because of higher air temperatures between the coils.

Rates of heat transmission obtained by tests for cast iron radiators and pipe coils are: (Steam 215°, room air 70°).

Cast Radiators	1-Col.	2-Col.	3-Col.	4-Col.
20 inches high	1.93	1.85	1.75	1.64
23 " "	1.89	1.80	1.70	1.59
26 " "	1.86	1.76	1.66	1.56
32 " "	1.79	1.69	1.59	1.49
38 " "	1.74	1.65	1.55	1.45
45 " "	....	1.60	1.50	1.40

Cast Wall Coils	
Heating surface 5 sq. ft., long side vertical	1.92
Heating surface 5 sq. ft., long side horizontal	2.11
Heating surface 7 sq. ft., long side vertical	1.70
Heating surface 7 sq. ft., long side horizontal	1.92
Heating surface 9 sq. ft., long side vertical	1.77
Heating surface 9 sq. ft., long side horizontal	1.98

Pipe Coils	
Single horizontal pipe	2.65
Single vertical pipe	2.55
Pipe coil 4 pipes high	2.48
Pipe coil 6 pipes high	2.30
Pipe coil 9 pipes high	2.12

93. Effect of Condition of Radiator Surface on the Transmission of Heat:—The efficiency of a radiator is affected by the condition of its outer surface. Painting, bronzing, shellacing or covering the radiator surface in any manner affects the rate of transmission of heat. A series of tests conducted by Prof. Allen at the University of Michigan, indicated that the ordinary bronzes of copper, zinc or aluminum caused a reduction in the efficiency below that of the ordinary rough surface of the radiator of 20 per cent., while white zinc paint, terra cotta enamel and white enamel gave the greatest efficiency, being slightly above that of the original surface. Numerous coats of paint, even as high as twelve, seemed to affect the efficiency in no appreciable manner, it being the last or outer coat that always determined at what rate the radiator would transmit its heat. Reference.—*Trans. A. S. H, & V. E.* The Effect of Painting Radiator Surfaces, J. R. Allen, Vol. XV, p. 229.

94. Effect of Housing on the Heat Transmission of Radiators:—Experiments were conducted by Prof. K. Brabbee of the Royal Technical Institute of Berlin, to determine a relation between the efficiencies of exposed and enclosed radiators. The results of these tests were reported in the Heating and Ventilating Magazine, May, 1914, and the following is a brief summary. No records were kept of volumes and temperatures of the air, these differing so greatly that the observations were of little value. The radiators used were two and three column, plain surface, ten sections, three inch centers. The two column radiators were 8.5 inches wide and the three column radiators were 9 inches wide.

Tests were first run with the radiators set in the ordinary way (2.5 inches from the wall, not enclosed and in

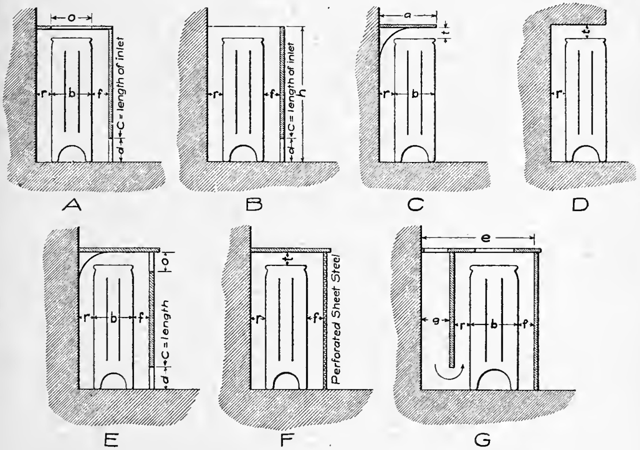


Fig. 82.

what is ordinarily called still air) giving results for K as follows: 49 in. 2 col. = 1.62; 24 in. 2 col. = 1.74; 50 in. 3 col. = 1.38; 26 in. 3 col. = 1.5. These values were then used as a basis of comparison for showing increased or reduced efficiencies of various housings. At the conclusion of the first series, tests were conducted upon the same radiators housed as shown in Fig. 82.

RESULTS FOUND.

A. The best spacing was found to be $r = f = 2.5$ inches, although K was about 8 per cent. less than in normal setting when thus enclosed. O should be at least the width and length of the radiator. When O was less than this amount K rapidly decreased. For open inlets $c =$ length of radiator and d min. $= 4$ inches. In such cases K was reduced 15 per cent. With inlet screened K was much less.

B. With $r = f = 2.5$ inches, the housing without top increased K as much as 12 per cent. because of the increased velocity of the air over the radiator due to the *chimney action.* The best results were obtained when the area of the inlet in square inches was approximately ten times the heating surface of the radiator in square feet. K increased by making enclosure higher than radiator.

C. Narrow shelves placed 3 inches or more above the radiators had little effect. Where a was such as to be flush with the front of the radiator and $t = 3$ inches K fell off about 5 per cent. On low radiators the loss was about 10 per cent. In either high or low radiators where $t = 4$ to 5 inches, K was about normal. Curved deflectors under shelves showed little or no gain in efficiency over the square corner.

D. Make $r = 2.5$ inches and $t = 3$ to 6 inches. Where $t = 3$ inches K was reduced 8 per cent. Where $t = 6$ inches, K was approximately normal. Side spacing had little or no effect.

E. A very inefficient form of housing even with d and o open slots. Under the very best conditions K was reduced 25 to 35 per cent.

F. With $r = f = t = 2.5$ inches, K was reduced about 20 per cent.

G. A very inefficient form of housing. K was reduced 30 to 40 per cent.

Tests of hot water radiators under the same conditions of housing verified the values found for steam. *A convenient way to apply the above* is to figure the square feet of radiator surface for normal setting and then multiply this amount by the following approximate values: A, 1.10; B, 1.00; C, 1.07; D, 1.10; E, 1.30; F, 1.20; G, 1.40.

In places where direct-indirect radiation is desired and no provision has been made for it in the building plan, Fig. 83 is suggested as a good substitute. The housing around

the radiator accelerates the draft and the damper arrangements give opportunity for all outside air, mixed outside and inside or all inside air at the discretion of the occupants.

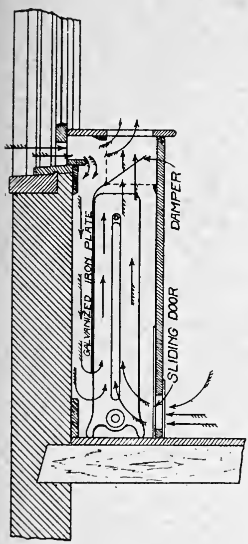


Fig. 83.

95. Amount of surface in Radiators:—Table XIII gives according to the columns and heights, the number of square feet of radiation surface per section in cast and pressed radiators. This table presents approximate values in very compact form from extended tables in the manufacturers' catalogs. An approximate rule supplementing this table and giving, to a very fair degree of accuracy, the square feet of surface in any standard radiator section, is as follows: *multiply the height of the sections in inches by the number of columns and divide by the constant 20; the result is the square feet of radiating surface per section.* The rule applies with least accuracy to one column radiators.

96. Pipe Fittings:—*Common and special.*—Pipes of standard diameters and random lengths are made from both wrought iron and steel. These pipes are cut, threaded with standard threads and connected with standard malleable or cast iron fittings to form any desired combination. Wrought iron pipe is considered by some to be more durable than steel pipe for general service but because of less first cost steel is more frequently employed. For exact diameters, surfaces, etc., see Table 29, Appendix.

Lengths of pipe are connected in straight runs by *unions* and *couplings.* Unions are threaded right-hand, and right-and-left. As distinguished from the right union the right-and-left has one end tapped right hand and the other left hand and connects between sections of a straight run already laid. Flanged couplings (usually packed between the flanges) are generally employed in connecting large sized pipes, and in addition are used on any sized pipe in places where sections may need fre-

TABLE XIII.

Dimensions and heating surfaces of radiators, per section.

Type of radiator	Max. spread of legs in inches	Ave. length per section in inches	Square feet of surface for over-all heights										
			45″	38″	32″	26″	23″	22″	20″	18″	17″	16″	14″
1 Col. C. I.	5½	2½	----	3	2½	2	1⅔	----	1½	1⅓	----	----	----
2 Col. C. I.	8½	2½	5	4	3⅓	2⅔	2⅓	2¼	2	1¾	----	1½	----
3 Col. C. I.	10	2½	6	5	4½	3¾	3¼	3	2¾	2¼	----	----	----
4 Col. C. O.	11¼	3	10	8	6½	5	4¼	4	3½	3	----	----	----
Flue wide	12½	3	----	----	----	----	----	----	6	5⅓	4¾	4⅔	4
Flue narrow	8½	3	----	7	5¾	4½	----	----	3¼	----	----	----	----
1 Col. press	5¼	2	----	3	2½	2	----	1⅔	----	1⅓	----	----	1
3 Col. press	8¾	2	6	5	4½	3¾	----	3	----	2¼	----	----	1.5
4 Col. press	12	2	----	----	----	4½	----	3¾	----	3	----	----	2¼

Wall rad. C. I. Thick. 3″	A. R. U. S.	13¼″x29⅛″ 14⅛″x29¼″	9 sq. ft.	13¼″x22″ 14⅛″x22⅞″	7 sq. ft.	13¼″x16¾″ 14⅛″x16½″	5 sq. ft.

quent removal for repairs or inspection. *Elbows* (usually called ells) change the direction of any run through 90°. See double pitch ells, Art 86. *Tees* are used where branches leave a straight run at 90°. They are sometimes made with a 45° instead of a 90° branch and are called *laterals* or *Y* fittings. Couplings, elbows, tees and laterals are made with varying inlet sizes. These are called *reducing couplings, reducing ells, reducing tees,* etc., and are specified as follows: state the sizes of the straight run, large and small, and second state the size of the branch. For illustration, a 2″ x 1½″ x 1″ reducing tee will change the size of the straight run from 2″ to 1½″ and give a branch of 1″. Where branches are made for water heating they should be so formed as to give a free and easy movement to the water. In such cases

it is desirable to use pipe bends having a radius of three to five pipe diameters, instead of the common elbow. In all cases pipe ends should be carefully reamed before assembling to remove the burr left by the cutter. This is most important in water heating as the burr on small pipes is sometimes heavy enough to reduce the area of the pipe by one-half, thus creating serious eddy currents and increasing the friction.

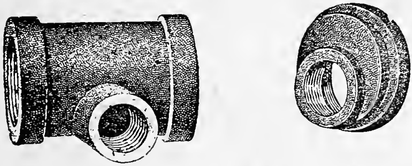


Fig. 84.

Eccentric reducing fittings (Fig. 84) are often of value in avoiding water pockets in steam lines. These should always be used on horizontal steam mains (Fig. 70) when reductions are made from one size to another. *Bushings* should not be used as reducing fittings for water lines because of the restriction to flow due to the square end of the bushing.

Valves are of two general types, *globe* and *gate*. Globe valves are installed on steam lines but they should not be used on horizontal steam mains where the seat will cause a water pocket and hinder drainage. Gate valves offer an unobstructed passage for both steam and water. They are recommended on all water lines and are being increasingly used on steam lines. Globe valves, however, are less expensive and are more easily repaired. The best type of globe valve has a renewable composition seat. Fig. 85 shows sections of each type.

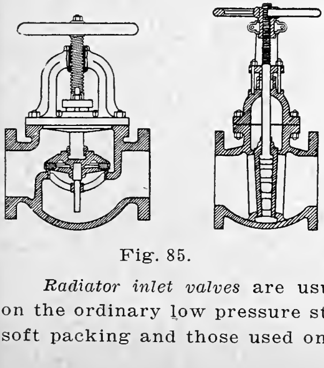


Fig. 85.

Radiator inlet valves are usually *angle* type. Those used on the ordinary low pressure steam systems are packed with soft packing and those used on systems which are occasion-

ally under partial vacuum are necessarily of the *packless* type. Those used on atmospheric and vacuum systems generally have graduated control. Fig. 86 shows several models of these valves. Water radiator valves are of the *quick opening* or *butterfly type,* opening and closing with a quarter turn of the handle and having a small hole through the valve to permit just enough leakage when closed to keep the radiator from freezing. *For radiator return valves to be used on mechanical vacuum systems, See Chapter IX.*

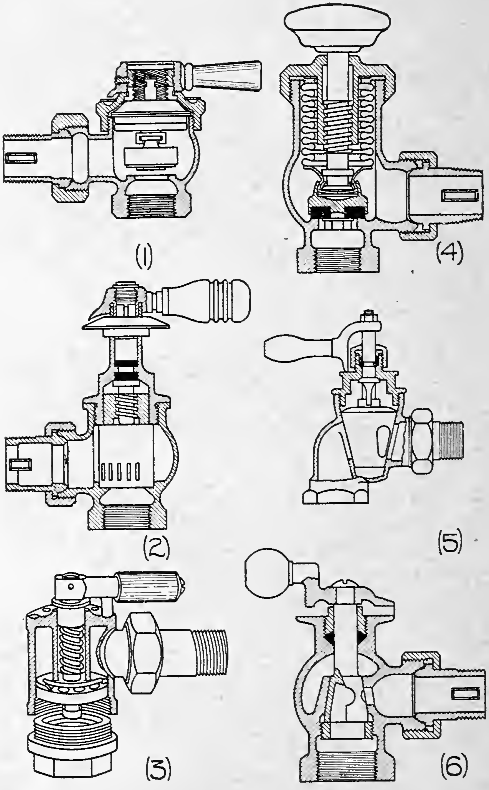
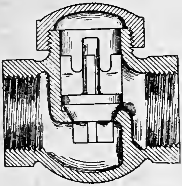


Fig. 86.

Check valves are of two kinds, *swing* and *lift*. They are not needed on the ordinary low pressure gravity water or steam systems, but where used swing checks should be specified rather than lift checks, for the former operate at less differential pressure and offer much less resistance to the passage of water and steam. Fig. 87 shows a section of each.



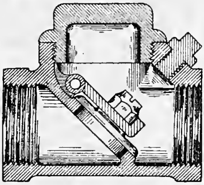


Fig. 87.

Air valves serve a most important function in heating systems. Air is constantly accumulating in the radiator and its frequent or automatic removal becomes necessary if all the radiating surfaces are to remain effective. For this purpose small hand operated valves or *compression* cocks, Fig. 88, are inserted near the top of the end section in all hot water radiators, and automatic valves are inserted at one-half to two-thirds the height of the last section on

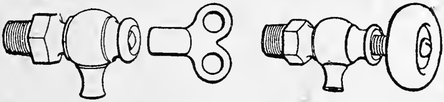


Fig. 88.

steam radiators. Air valves are not essential to two-pipe steam systems and are sometimes omitted. They are not needed on vapor systems and are always omitted. Fig. 89 shows a common type of automatic air valve using the principle of the expansion stem. As long as air is in contact with the stem it remains contracted and the needle valve is open for air release. When steam enters the valve it surrounds the stem and expands it sufficiently to close the needle valve and prevent steam loss. Fig. 90 operates on the principle of the evaporation of a volatile liquid in a closed container. The composition of this liquid is such that

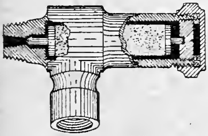


Fig. 89.

it evaporates at the steam temperature and causes a deflection of the base of the container sufficiently to move the valve pin and close the valve. Air temperatures being less than steam temperatures, the reverse action takes place when air collects around the stem and the valve opens for air release. Water jetting, which is frequently found with air valves, is eliminated by the floatation of the container which is free to lift and ride the water that collects in the float chamber. A modification of this valve (Fig. 91) has, in addition to the thermal features just mentioned, an atmospheric air pressure feature which permits its use on vacuum systems. As the pressure is reduced in the radiator, atmospheric air presses upward against diaphragm 1 and forces the pin valve against its seat. It will be seen that when the pressure within the radiator is less than atmospheric the differential pressure closes the valve and keeps air out. When the pressure within the radiator is

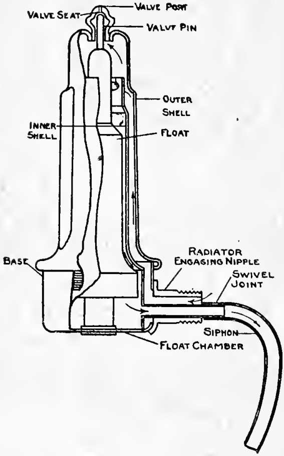


Fig. 90.

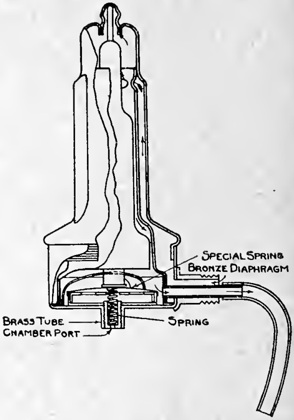


Fig. 91.

slightly above atmospheric the valve stands open except at the times when all the air is exhausted and the steam holds the valve closed through the expansion member. By the combined action of both expansion members air may be released continuously but it can not reenter. Valves operating on either the thermal or differential pressure principle

or both may be had for quick venting on mains, coils and other parts of a heating system. Figs. 92 and 93 operate on the principle of the difference of expansion between two dissimilar metals. In the first one the expansion member is made of two strips of dissimilar metals brazed together in the form of a loop. These metals expand at different rates under changes of heat and cause endwise movement of the

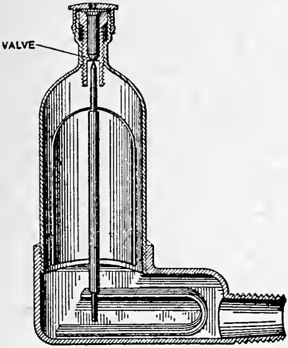


Fig. 92.

Fig. 93.

valve rod, thus opening or closing the valve. The hollow float serves the purpose of catching any sudden surge of water and avoids flooding. The second one employs a long central tube which carries at the top, the valve seat of the needle valve. The needle itself is carried by the two side rods. As long as the air flows up through the central pipe, the needle valve will remain open, but when steam enters the tube it expands and carries the valve seat upward against the needle, thus closing the valve. The size and strength of parts make this form a very reliable one. *For air line valves to be used on mechanical air line systems, see Chapter IX.*

97. Expansion Tank:—The expansion tank is a necessity in all atmospheric hot water systems. Its function is to serve as a supply tank for the system and also as a take-up for the excess volume due to the heating of the water. Fig. 94 shows a typical cylindrical galvanized tank supplied in capacities of 8, 10, 15, 20, 26, 32, 42, 66, 82 and 100 gallons; the average size, 16 in. diam. x 30 in. high is rated for approximately 1000 square feet of radiation including mains.

Fig. 95 is an automatic, self-filling, copper lined tank approximately 20 in. x 9 in. x 10 in. and is supplied for systems up to 2000 square feet capacity. The galvanized tank is tapped 1-inch for the overflow and expansion pipes, and the automatic tank is tapped ⅜-inch supply, 1¼-inch expansion and 1½-inch overflow. The expansion tank is often

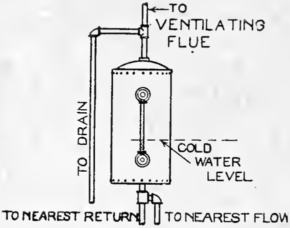


Fig. 94

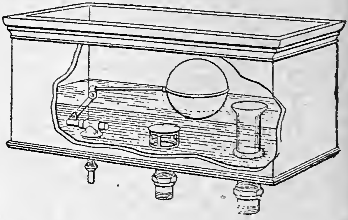


Fig. 95.

located in the bath room or a closet near the bath room and its overflow connected to the proper drainage. It should be set at least two feet above the highest radiator. The connection between this tank and the heating system is often by a branch from the nearest radiator riser. The best connection is by an independent riser from the basement return main. The capacity of the tank for any system up to 1500 square feet may be obtained by the approximate rule: *divide the total radiation by 40 to obtain the capacity of the tank in gallons.*

98. Fire Coils or Water Backs for Hot Water Supply:— Pipe coils or cast iron water chambers (water backs) may be installed in the combustion chamber of any furnace, hot water or steam plant for heating the domestic hot water supply. Care must be exercised in installing these fixtures to see that there is an up-flow for the water from the point of entering to the point of leaving the fire box. *Soft water should be used in these water systems wherever possible because of the lime and other deposits thrown off from the hard water.* If it becomes necessary to circulate hard water the coils should be examined at least once each year to see that they are not filled with lime. Lime deposits cut down the heat transmission, cause the pipes to burn and endanger the plant in making it more liable to explosion. In many plants the heating surface on these coils is excessive. On cold days under heavy fire the water in the tank is maintained at the

boiling point when lower temperatures would be more satisfactory. This condition may be corrected by attaching water connections to the circulating pipes outside the fire box and running these leads to a hot water radiator where heat is most needed. Each lead to the radiator and the flow pipe to the hot water tank should be valved with gate valves so as to regulate the circulation in each line.

99. Corrosion of Pipes:—Much has been said and written about the internal wastage or wearing away of steel and wrought iron pipes conveying hot water, but owing to the fact that such a long time is necessary for a comparative test and surrounding conditions are so changeable, no authoritative data have yet been found to prove conclusively to pipe users that either of the two (steel or iron) is longer lived than the other. Most of the pipe now used in the country is of mild steel, probably because of the fact that this pipe can be manufactured and marketed at a lower price. Nevertheless if it may be shown by any conclusive proof that wrought iron pipe is more durable the price would be a secondary feature in the purchase. One of the most convincing papers on this subject yet presented to the engineering profession is found in the *Trans. A. S. H. & V. E.*, Vol. 24, p. 217, by F. N. Spellor and R. G. Knowland. A copy of the paper is also found in Technical Paper 236, Department of the Interior, Bureau of Mines.

CHAPTER VIII.

HOT WATER AND STEAM HEATING.

PRINCIPLES OF THE DESIGN, WITH APPLICATION.

100. Selecting Boilers for Capacity:—To determine the necessary boiler capacity for a given installation, find the theoretical grate surface to supply the calculated heat loss plus 20 to 30 per cent. to cover that lost from the mains and risers, and select a boiler having at least this amount of grate from the catalog data representing the type of boiler desired. Current practice adds 25 to 50 per cent. to the theoretical grate as a safety margin. *Rule.—To find the theoretical grate surface in square feet, divide the total B. t. u. required per hour for maximum heating service by the product of the pounds of coal estimated per square foot of grate surface per hour (rate of combustion), the efficiency of the furnace and the heat value of the fuel in B. t. u. per pound.* (See Equation 46).

The following rates of combustion may be used for internally fired heating boilers:

Sq. ft. of grate	4–6	6–10	10–18	18–30
Lbs. coal per sq. ft. grate per hour	5	6	8	10

Boilers with constant attendance, such as power boilers, may have a higher rate of combustion.

Catalog ratings are usually obtained from test data taken when the boilers are burning anthracite coal. Where boilers are to be used with soft coals the 50 per cent. additional capacity mentioned above had best be taken because of the larger volume needed per pound of coal and because of the sooty nature of the coals.

APPLICATION.—Assume a total building heat loss (including mains and risers) of 150000 B. t. u. per hour; soft coal 13000 B. t. u. per lb.; 5 pounds coal per sq. ft. of grate per hour; and boiler efficiency 60 per cent.; then from Equation

46, G. A. = 550 sq. in. Add 50 per cent. = 825 sq. in. = 5.7 sq. ft. From the Ideal Fitter this gives an S-25-5. Sectional boiler with a hard coal rating of 1100 sq. ft. of heating surface.

101. Calculation of Radiator Surface—Direct Radiation: —In designing a hot water or steam system, the first important item to be determined is the square feet of radiation to be installed in each room. Nearly all other items, such as pipe sizes, grate area, boiler size, etc., are estimated with relation to the radiation supplied. The correct determination then, of the square feet of radiation in these systems is all important. The general equation used to obtain the square feet of radiation for any room is:

$$R = \frac{\text{total B. t. u. lost from the room per hour}}{K\ (\text{av. temp. diff. between inside and outside of rad.})}$$

Rule.—To find the square feet of radiation for any room divide the calculated heat loss in B. t. u. per hour by the quantity K times the difference in average temperatures between the inside and outside of the radiator.

Expressed in symbols

$$R_w = \frac{H \text{ (or } H')}{K\left(\dfrac{t_a + t_b}{2} - \dfrac{t + t_o}{2}\right)} \tag{49}$$

$$R_s = \frac{H \text{ (or } H')}{K\left(t_s - \dfrac{t + t_o}{2}\right)} \tag{50}$$

Where R_w and R_s = sq. ft. radiation required for water and steam heating, t_a and t_b = water temperatures entering and leaving radiators, t and t_o = temperatures of air passing over radiator and t_s = temperature of the steam. In ordinary direct radiation calculations the term $[(t + t_o) \div 2]$ is usually taken 70.

In Art. 92, the *rate of transmission K*, (Amount of heat transmitted through one square foot of surface per hour per degree difference in temperature between the inside and the outside of the radiator) obtained from tests, varies inversely with both the height and the width of the radiator, being as high as 1.93 for low 1-column and as low as 1.40 for high

4-column radiators. For extreme accuracy these values may be used. For ordinary service, however, they may be summarized with fair accuracy into:

low radiators	—16 to 23	inches	—1.8
medium "	—23 to 32	"	—1.7
high "	—32 to 45	"	—1.6

All applications in this book will be taken 1.7.

With hot water as the heating medium the temperatures within the radiator for the open tank system are about as follows: entering the radiator 180°; leaving the radiator 160°; average temperature on the water side 170°. To find the amount of hot water radiation for any other average temperature of the water, substitute the desired average temperature in the place of 170. The maximum drop in temperature as the water passes through the heater will seldom be more than 20 degrees even under severe conditions. The temperature of the entering water may be assumed as high as 212° if it is considered necessary in which case each square foot of surface becomes more efficient and the total radiation in the room may be reduced. Since radiators become less efficient from continued use, it is best to design a system with lower temperatures as stated and under stress of conditions the capacity may be increased by raising the flow temperature to the boiling point. With a room temperature of 70°, a 26-inch 2-col. or 3-col. hot water radiator will give off $1.7 \times (170\text{-}70) = 170$ B. t. u. per square foot per hour and the amount of radiation is:

For *hot water, open tank direct radiation* as usually applied

$$R_w = \frac{H}{1.7\,(170 - 70)} = \frac{H}{170} = .006\,H \qquad (51)$$

For the Honeywell system and others maintaining pressures above atmospheric, use higher water temperatures in the general equation. For example, suppose these temperatures are entering at 220° and leaving 200°, we have

$$R_w = \frac{H}{1.7\left(\dfrac{220 + 200}{2} - 70\right)} = \frac{H}{238} = .0042\,H \qquad (52)$$

A steam system may be installed to work at any pressure from a partial vacuum of, say 10 pounds absolute, to as

high a pressure as 75 pounds absolute. To calculate the proper radiation for any of these conditions use Equation 50 and substitute the proper steam temperature.

The temperature within a steam radiator carrying steam at pressures varying between 0 and 3 pounds gage may be taken 220°; then the total transmission for this radiator will be $1.7 \times (220 - 70) = 255$ B. t. u. per square foot per hour, and the amount of radiation

For *Steam, gravity direct radiation* as usually applied is

$$R_s = \frac{H}{1.7\,(220 - 70)} = \frac{H}{255}, \text{ safe value } = \frac{H}{250} = .004\,H \quad (53)$$

It will be seen from Equations 51 and 53 that $R_w = 1.5\,R_s$. This ratio is frequently used 1.6. (See also Art. 137 for logarithmic equation).

For *atmospheric and vapor systems with individual traps* on the return end of each radiator

$$R_x = \frac{H}{1.7\,(212 - 70)} = \frac{H}{241}, \text{ safe value } = \frac{H}{240} = .00417\,H \quad (54)$$

For *atmospheric and vapor systems with open return* and 20 per cent. excess radiation for cooling surface

$$R_y = \frac{H \times 1.2}{1.7\,(212 - 70)} = \frac{H}{200} = .005H \quad (55)$$

Note.—Equations 51 to 55 will meet average conditions. If for high radiators, low radiators, wall or pipe coils it is considered necessary to be more specific, use the values K given in Art. 92.

Application.—Referring to the standard room with $H = 15267$, Art. 39, the equations quoted above give:

(51) hot water, open tank 91 sq. ft.
(52) hot water, closed tank 64 sq. ft.
(53) Steam, 0 to 3 lbs. 61 sq. ft.
(54) Steam, vapor, closed returns 64 sq. ft.
(55) Steam, vapor, open returns 76 sq. ft.

Assuming a 26-inch 3-col. type cast radiator, we have as follows:

for (51), 24 sections, radiator 60 inches long.
for (52), (53), (54), 17 sections, radiator 42.5 inches long.
for (55), 21 sections, radiator 52.5 inches long.

Many *empirical equations* and rules have been devised (based somewhat upon the rational Equations 49-55) in an attempt to simplify calculation, but their applications are

untrustworthy unless used with that discretion which comes with years of practical experience. The reason why such empirical rules often give erroneous results is because of the fact that nothing is said concerning exposure and *equivalent wall,* such as floors and ceilings. Some of these equations and their applications to the "Study" Art. 62, where $G = 48$, $W = 192$ and $C = 1900$ are:

$$\text{(a)} \quad R_w = \frac{G}{2} + \frac{W}{10} + \frac{C}{60} = 24 + 19.2 + 31.6 = 74.8 \text{ sq. ft.}$$

$$\text{(b)} \quad R_w = G + \frac{W}{20} + \frac{C}{100} = 48 + 9.6 + 19 = 76.6 \text{ sq. ft.}$$

$$\text{(c)} \quad R_w = \frac{3G}{4} + \frac{W}{10} + \frac{C}{100} = 36 + 19.2 + 19 = 74.2 \text{ sq. ft.}$$

$$\text{(d)} \quad R_s = \frac{G}{2} + \frac{W}{10} + \frac{C}{200} = 24 + 19.2 + 9.5 = 52.7 \text{ sq. ft.}$$

$$\text{(e)} \quad R_s = \frac{2}{3}\left(G + \frac{W}{20} + \frac{C}{100}\right) = \frac{2}{3}(48 + 9.6 + 19) = 51 \text{ sq. ft.}$$

$$\text{(f)} \quad R_s = \frac{G}{2} + \frac{W}{20} + \frac{C}{200} = 24 + 9.6 + 9.5 = 43.1 \text{ sq. ft.}$$

Checking these Equations 51 and 53 we have $R_w = 76$ sq. ft. and $R_s = 52$ sq. ft.

102. Direct-Indirect Radiation:—This system of heating is used in some homes and in many moderate sized school buildings. It has the simplicity of direct radiation with certain ventilating possibilities which may be had at a reasonable first cost. In discussing direct-indirect heating, however, it should be remembered that the ventilating feature of the system is very erratic, having a tendency to move too much air when the wind pressure is against that side of the building where the radiator is located and too little air when the direction of air movement is reversed. The requir ment of a constant supply of 1800 cubic feet of air per person can not be maintained by convection processes solely. The reason for this is the low velocity of the air entering the building (even in some cases a reversal of movement) at times when the air pressure is reversed. In order to keep from over heating the room with too much direct-indirect radiation (found necessary in keeping up the air supply) some reduction must be made from the usual requirement of

ventilation when designing this type of system, say to 1200 or 1500 cubic feet. *Direct-indirect radiation should always be used in connection with inner wall ventilating stacks and preferably those fitted with aspirating coils.* Such stacks have a pull on the room air and overcome to a certain extent the back draft of the room air over the radiator.

In school house heating, direct-indirect radiation is usually installed in connection with direct radiation. The two kinds may be assembled in each radiator or certain radiators may be all direct and others all direct-indirect as preferred. Ten to twelve sections of the standard radiator are usually connected to one wall box. *Wall boxes* are made in varying sizes; one standard form being sold in three sizes —8x24, 8x30 and 8x36 inches, having approximately 100, 125 and 160 square inches net area respectively. *High radiators should be used because of the chimney effect in overcoming back drafts. Low pressure hot water, vacuum or atmospheric steam systems should be used with caution because of the danger of freezing.*

In estimating direct-indirect radiation with the accompanying duct sizes, it may be done by either one of two methods: *first*, estimate the direct-indirect radiation to supply the necessary heat to warm the amount of ventilating air desired and add sufficient direct radiation to make up the balance of heat for the calculated heat loss, H; *second*, estimate the direct radiation necessary to supply H and add 50 per cent. for indirect heat given to the entering air, then enclose and connect to wall boxes the necessary radiation for direct-indirect work.

Application.—Assume a standard recitation room in a school building having 13″ brick walls; the room to be 24 ft. x 30 ft. one side and one end exposed, window area = one-sixth the floor area, 12 ft. ceiling, H = 50000 B. t. u., and arrangement of seating (excepting 8 feet across the front of the room reserved for instructional purposes) 15 square feet of floor space per pupil.

Analysis.—Number of pupils 35. Amount of air required for ventilation (say 1400 cu. ft. per pupil) = 49000 cu. ft. per hour. Select medium sized wall box 125 sq. in. net wind area. With favorable conditions we may expect 1 to 2 cu. ft. air per min. per sq. in. net wall box area (air velocity 2.5 to 5 f. p. s.) Call this 1.5 cu. ft. We have $125 \times 1.5 \times 60 = 11250$ cu. ft. per hour per wall box. $11250 \div 1400 = 8$ pupils

supplied. Four wall boxes 8 in. x 30 in. will approximately supply all the pupils at the rate of 1400 cu. ft. per person. These theoretically should give $11250 \times 4 = 45000$ cu. ft. air per hour. Since this number of radiators makes a good division for the room, the same number of radiators may be used. In the direct-indirect arrangement just mentioned, with average air velocities, air temperatures may be raised from zero to 125° and each sq. ft. of included radiation will give off approximately 375 B. t. u. per hour (when the outdoor air is below zero it will be advisable to recirculate part or all of the air). The heat given to the air will be $[45000 \times (125 - 0)] \div 55 = 102272$ B. t. u. and the radiation will be $102272 \div 375 = 273$ sq. ft. = four radiators 68 sq. ft. each (approximately 160 cu. ft. of air per sq. ft. of radiator surface). In all probability these would be taken 12 sec. 38-in. 3 col. 60 sq. ft. With 60 sq. ft. in each radiator the total heat given off to the air in the room will be approximately $4 \times 60 \times 375 = 90000$ B. t. u. Of this amount of heat 56 per cent. (50404 B. t. u.) is used to raise the temperature of the air from zero to 70° and 44 per cent. (39600 B. t. u.) is used to raise it from 70° to 125°. This latter amount will be given off to the room air and is a credit to the heat loss H. $50000 - 39600 = 10400$ B. t. u. to be supplied by direct radiation. $10400 \div 250 = 41.6$ sq. ft. = 8.3 sections, 38-in. 3-col. radiation. Assuming this to be 8 sections and divided equally among the radiators we have four 38-in. 3-col. radiators each consisting of 12 sec. direct-indirect and 2 sections direct radiation. This would be considered a fairly satisfactory arrangement. The direct-indirect radiation should be installed so as to operate as such on outside air or as direct on recirculated air if desired.

Under the second method suggested find $50000 \div 250 = 200$ sq. ft. of direct radiation to offset H. Add 50 per cent. = 300 sq. ft. total = four 38-in. 3-col. radiators each 15 sec. divided 12 sec. direct-indirect and 3 sec. direct.

Comparing the amount of radiation obtained by the first method with the amount required if heated by direct radiation, we have

	direct radiation	200 *sq. ft.*	= 1.00
Combined	*direct-indirect radiation*	240 *sq. ft.*	= 1.20
	direct radiation	40 *sq. ft.*	= .20

103. Gravity Indirect Radiation:—This provides one of the most satisfactory systems of heating. In all essentials it compares with the furnace system, with the furnace replaced by individual steam or hot water radiators. (See Fig. 96). It is an improvement over the direct-indirect system in that there is a fairly constant air movement to the room. Radiators of either the extended pin or ribbed type are used. Some of the standard sizes are given in Table XIV.

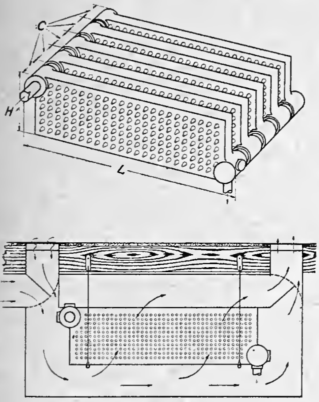


Fig. 96.

The indirect radiator should be set 20 to 24 inches above the water line of the boiler. There should be a clearance of 10 inches at the top and 8 inches at the bottom between the casing and the radiator, for cold air and warm air chambers, but the casing on the sides and ends should be close to the radiator. The radiators are suspended from the joists and connected to the steam and return mains in such a way as to permit free expansion and contraction. All pipes must be graded for free drainage to the boiler.

TABLE XIV.

	Gold Pin			Sanitary School Pin		Perfection Pin	Pin Indirect		
Sq. ft. H. S. per sec.	12	15	20	15	20	10	10	15	20
L	36	36	36	36⅛	36⅛	36¼	36¼	36⅝	36
H	9	11½	15¼	11½	15¼	9$\frac{13}{16}$	8⅝	11⅝	14¾
C	3¼	3¼	3¼	4	4	2¾	3	3	3½

Pipe sizes may be the same as those used on any two-pipe radiator having equivalent condensation. Low radiator sections (h = 6 to 10 inches) are recommended for residences and offices. High radiators (h = 10 to 15 inches) are recommended for schools.

To determine the *amount of air circulated per hour,* read Arts. 49-51. In residences and offices, air as a heat carrier will be sufficient. In schools and auditoriums there will be an excess of air for ventilation. Where this is true the temperature of the air leaving the radiator should be correspondingly below what it would be if only heating were considered. (See Art. 52).

The *lowest temperature of the entering air* may be taken zero. At lower temperatures part or all of the air should be recirculated. The *temperature of the air leaving the radiator* depends upon the velocity of movement over the radiator.

Table XV, from experiments by J. R. Allen, cols. 2 and 3, gives air temperature rise in passing over the radiator.

TABLE XV.

Cubic feet of air passing per sq. ft. of radiation per hour	Rise in temperature of the air		Pounds of steam Condensed per sq. ft. of radiation		B. t. u. transmitted per sq. ft. of radiation per degree difference in temp. between steam and air	
	Standard pin	Long pin	Standard pin	Long pin	Standard pin	Long pin
50	147	140	0.125	0.150	0.80	0.95
75	143	137	0.170	0.210	1.17	1.27
100	140	135	0.240	0.260	1.51	1.60
125	138	132	0.295	0.310	1.85	1.90
150	135	129	0.355	0.360	2.22	2.20
175	132	126	0.410	0.405	2.57	2.47
200	130	123	0.470	0.450	2.90	2.72
225	127	120	0.530	0.490	3.25	3.00
250	123	118	0.585	0.530	3.60	3.20
275	121	115	0.645	0.570	3.90	3.40
300	119	112	0.700	0.610	4.22	3.60

In the design of indirect heating systems there are certain approximate values which may be recommended as representing fairly standard practice. These values which follow may be used in connection with Table XV.

Cu. ft. of air per sq. ft. of radiation
(residence) ....steam 150 water 100
K for residence heating 2.2
Cu. ft. of air per sq. ft. of radiation
(schools) .... " 200 " 133
K for school heating 2.6
Temperature of air entering radiator...... zero
Temperature of air leaving radiator
(residence) .... 100, 125 and 150

Sq. ft. of radiation determined by Equation 56 or 57

$$R_s = \frac{H'}{K\left(t_s - \frac{t + t_o}{2}\right)} \tag{56}$$

$$R_w = \frac{H'}{K\left(\frac{t_a + t_b}{2} - \frac{t + t_o}{2}\right)} \tag{57}$$

Where terms are as stated in Art. 101.

Sq. in. of flue area per sq. ft. of rad.		Steam	Water
Height between c. of rad. and c. of reg.	5 ft.	2.00	1.33
" " " " "	10 ft.	1.40	.90
" " " " "	20 ft.	1.00	.66

Select type of radiator from catalog data.

Indirect radiators are usually arranged to permit recirculation of the air from the house when desired. For other information on recirculating ducts, registers, etc., see furnace heating.

Application 1.—In Art. 62, the Living Room ($H = 15267$) and Chamber 1 ($H = 10583$) are to be supplied with indirect steam heat, design the heaters and heat lines.

Solution.—Assume $t_o = O$; $t = 125$ and $t_s = 220$; then for the Living Room

$$Q \text{ (Eq. 33)} = \frac{55 \times 15267}{125 - 70} = 15267$$

$$H' \text{ (Eq. 30)} = 15267 + \frac{15267 \times (70 - 0)}{55} = 34698$$

$$R_s \text{ (Eq. 56)} = \frac{34698}{2.2\left(220 - \frac{125 + 0}{2}\right)} = 100 \text{ sq. ft.}$$

Efficiency of radiator $= 2.2\ (220 - 62.5) = 346.5$ B. t. u

Amount of circulating air $= 100 \times 150 = 15000$ cu. ft.

Compare this with calculated value of Q.

Check amount of indirect radiation with direct radiation, Art. 101, Eq. 53. This shows an increase of $(100 - 61) \div 61 = 64$ per cent. above the calculated direct radiation for the same room.

Square inches warm air duct area = 2 × 100 = 200.

Square inches cold air duct area = 200 × .8 = 160.

For *Chamber 1*, with temperatures as before

$$Q = \frac{55 \times 10583}{125 - 70} = 10583$$

$$H' = 10583 + \frac{10583 \times (70 - 0)}{55} = 24143$$

$$R_s = \frac{24143}{2.2\left(220 - \frac{125 + 0}{2}\right)} = 70 \text{ sq. ft.}$$

Efficiency of radiation = 346.5 B. t. u.

Check amount of circulating air, 70 × 150 = 10500 cu. ft.

Compare this with Q.

Check indirect radiation with direct radiation, 66 per cent. increase.

Square inches of warm air duct area = 1.25 × 70 = 88

Square inches of cold air duct area = 88 × .8 = 70

Application 2.—In Art. 102, a school room 24 x 30 ft. has a heat loss of 50000 B. t. u. and has 35 pupils. It is required that this room be heated by indirect radiation, design the heaters and heat lines.

Solution.—$Q = 35 \times 1800 = 63000$; $t_o = 0$; $t_s = 227$.

$$t = \frac{50000 \times 55}{63000} + 70 = 113.6 \text{ say } 114°.$$

$$H' = 50000 + 63000 \times 1.27 = 130180.$$

$$R_s = \frac{130180}{2.6\left(227 - \frac{114 + 0}{2}\right)} = 295 \text{ sq. ft.}$$

Efficiency of radiator = 442 B. t. u.

Check amount of circulating air, 63000 ÷ 295 = 213 cu. ft. per sq. ft. radiation.

Check amount of indirect radiation with direct radiation for the same room and find direct = 200, indirect = 295; approximately 1.00 : 1.50.

104. Aspirating Coils:—For the most efficient service, direct-indirect and indirect heating should be accompanied by a *positive withdrawal* of the air from the rooms through ventilating ducts. This is true especially in the heating of school buildings. Individual electric driven fans may be

housed in the vent ducts or the vents may be gathered together in the attic and housed in around one exhaust fan, but these plans require extra care in installing and are expensive in first cost. Furthermore, in many places electric power can not be had. In such places indirect radiation (aspirating coils) may be placed in the vents as shown by Fig. 97 and the heat given off will produce convection air currents which insure a positive withdrawal of the room air. This is not an efficient method of producing draft, in fact any other workable method should be employed where possible.

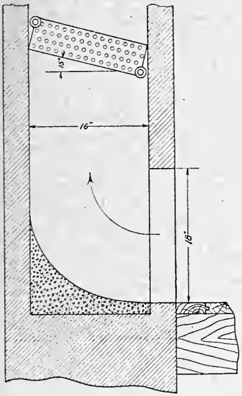


Fig. 97.

In installing aspirating coils they should each be piped direct from the boiler room. The valves should be located in the boiler room and should be under the control of the boiler attendant. When the rooms are not occupied, steam should be cut out of the coils and the vent dampers closed to avoid depleting the room of warm air. When coils are cut off they should be drained to avoid freezing.

The amount of radiation to install in each vent flue varies in different localities. Fairly satisfactory results seem to be obtained with two vents to each room (25 x 30 x 12 ft.) and 30 to 40 square feet of cast iron indirect radiation in each vent. For sizes of vent ducts above heaters see corresponding sizes in furnace heating.

105. Greenhouse Heating:—In estimating greenhouse radiation the problems are essentially different from those in ordinary house radiation. In greenhouses glass surface is large, wall surface is small and air circulation is comparatively less. The rational equation for heat loss, therefore, has less to do with volume and except in unusual cases may be considered to have but two terms—glass and wall. Where volume is accounted for, calculate H as in Chap. III. Instead of ordinary cast iron radiation, the radiating surfaces are wrought iron or steel pipes 1¼-, 1½- or 2-inch diameter (or

cast pipes 2½- to 4-inch diameter) assembled as coils with manifold headers. The values of K for these coils may be found in Art. 92. Although test values run as high as 2.65 for single horizontal pipes it is a safe plan because of the dirt deposits on and in the pipes, to allow an average value of not to exceed 2.2 for all wrought iron or steel coils and 1.8 for all cast iron coils. Find H by Equation 26 (or 27), using only glass and wall equivalent and substitute in Equations 49 and 50. For all practical purposes $H = (G + .25\ Eq.\ W)\ 70$.

Assuming wrought or steel coils and zero weather

$$R_w = \frac{H}{2.2\ (170 - 70)} = .0045\ H \tag{58}$$

$$R_s = \frac{H}{2.2\ (220 - 70)} = .0030\ H \tag{59}$$

If the desired indoor temperature is other than 70°, this temperature should be substituted for 70 in the equation. Assuming $t' = 70$ we have $R_w = .32\ (G + .25\ Eq.\ W)$ and $R_s = .21\ (G + .25\ Eq.\ W)$; = *one square foot of H. W. radiation to 3.1 square feet of equivalent glass area and one square foot of steam radiation to 4.8 square feet of equivalent glass area.* Table XVI (Model Boiler Manual) shows the amount of surface for different interior temperatures and different temperatures of the heating medium.

TABLE XVI.

Temp. of air in Green-house	Temperature of water in heating pipes 140°	160°	180°	200°	Steam Three lbs. pressure	
	Square feet of glass and its equivalent proportioned to one square foot of surface in heating pipes or radiator					
40°	4.33	5.25	6.66	7.69	8.0	10.00
45°	3.63	4.65	5.55	6.66	7.5	8.50
50°	3.07	3.92	4.76	5.71	7.0	7.40
55°	2.63	3.39	4.16	5.00	6.5	6.60
60°	2.19	2.89	3.63	4.33	6.0	5.90
65°	1.86	2.53	3.22	3.84	5.5	5.20
70°	1.58	2.19	2.81	3.44	5.0	4.80
75°	1.37	1.92	2.50	3.07	4.5	4.30
80°	1.16	1.63	2.17	2.73	4.0	3.90
85°	.99	1.42	1.92	2.46	3.5	3.50

This table is computed for zero weather; for lower temperatures add 1½ per cent. for each degree below zero. The

last column was calculated from Equation 50 ($K = 2.2$) and added for purpose of comparison.

Empirical rules for greenhouse radiation are sometimes given in terms of the number of square feet of glass surface heated by one lineal foot of 1¼-inch pipe. One such rule is —"one foot of 1¼-inch pipe to every 2¼ square feet of glass, for steam; and one foot of 1¼-inch pipe to every 1⅓ square feet of glass, for hot water, when the interior of the house is 70° in zero weather." Great care should be exercised in rating and selecting the boilers and heaters. It is well to remember that the severe service demanded by a sudden change in the weather is much more difficult to meet in greenhouses than in ordinary structures, and that a liberal reserve in boiler capacity is highly desirable.

Both steam and hot water systems are in general use. Where continuous heat may be obtained throughout the night from a central plant a steam system is very desirable. In the isolated plant where the steam pressure drops during the night time a hot water system will give more satisfactory service in cold weather because it guarantees a better circulation of heat throughout the night.

The same rules apply in running the mains and risers as apply in the ordinary hot water and steam systems. In greenhouse work the head of water in a water system is necessarily very low and tends to make the circulation sluggish, but with sufficient pipe area to reduce the friction a hot water open tank system having a very low head may be made to work satisfactorily. In some houses the coils are run along the wall below the glass and supported on wall brackets; in others they are run underneath the benches and supported from the benches with hangers. In greenhouses with very large exposure there are sometimes required both wall and bench coils, also, a certain amount in

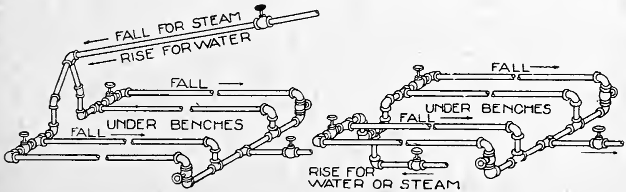


Fig. 98.

the center, 6 to 7 feet from the floor. In all of these piping layouts it is necessary that as much rise and fall be given to the pipes as possible. Fig. 98 shows two systems of pipe connections, one where the steam or flow enters the coils from above the benches and the other where it enters from below, the return in each case being at the lowest point. These bench coils could be run along the wall with equal satisfaction.

Application.—Given an even-span greenhouse 25 ft. wide, 100 ft. long and 5 ft. from ground to eaves of roof, having the slope of the roof with the horizontal 35°. The ends to be glass above the eaves line. What amount of hot water radiation with average water temperature 170°, interior temperature 70° and outside temperature 0°, and what amount of low pressure steam radiation should be installed?

Length of slope of roof $= 12.5 \div \cos. 35° = 15.25$. Area of glass $= 15.25 \times 100 \times 2 + 2 \times 12.5 \times 8.8 = 3270$ sq. ft. Area of wall $= 5 \times 100 \times 2 + 5 \times 25 \times 2 = 1250$ sq. ft. Glass equivalent $= 3270 + .25 \times 1250 = 3582.5$ sq. ft. $R_w = .32 \times 3582.5 = 1146$ sq. ft. $R_s = .21 \times 3582.5 = 752$ sq. ft. From Table XVI. $R_w = 3582.5 \div 2.81 = 1270$ sq. ft. $R_s = 3582.5 \div 5 = 716$ sq. ft. *Check with last column of Table XVI.

References.—*Jour. A. S. H. & V. E.* Heating a Conservatory and Greenhouse, July 1916, p. 29. *Metal Worker.* Design of Greenhouse Heating Plants, July 9, 1915, p. 44. Heating Equipment for Large Greenhouse, Jan. 1, 1915, p. 66. *Domestic Engineering.* The Hot Water Heating in Highland Park Greenhouse, Sept. 21, 1912, p. 292.

106. The Theoretical Determination of Pipe Sizes:—The theoretical determination of pipe sizes for small hot water and steam systems has always been more or less unsatisfactory because of the difficulty in estimating the friction offered by different combinations of piping. The following analysis is illustrative and does not account for friction.

Assume *a hot water system,* Figs. 101-104, having water temperatures entering and leaving the radiators 180° and 160° respectively. Since one pound of water in passing through each radiator gives off 20 B. t. u., the radiators in the Living Room (91 sq. ft.) and Chamber 2 (70 sq. ft.) will require 91 and 71 gallons of circulating water per hour (check the values), or approximately *one gallon of water per*

square foot of radiating surface per hour. This is a general statement which will be true for any low pressure hot water system with 20 degrees temperature drop. With the amount of water required per hour obtain the velocity due to the unbalanced columns and find by division the area of the pipe. Assume the radiator in the Living Room to have a 5-ft. static head and that in Chamber 2 a 15-ft. head. Having the water temperature in the flow risers 180° and in the return risers 160° (good values in practice), the heated water in the flow risers weighs 60.5567 pounds per cubic foot, while that in the return risers weighs 60.9697 pounds per cubic foot. The motive force is $f = g \times \dfrac{W - W'}{W + W'}$, where g is the acceleration due to gravity, W is the specific gravity (weight) of the cooler column and W' is the specific gravity (weight) of the warmer column. Substitute f for g in the velocity equation and obtain

$$v = \sqrt{2\,fh} = \sqrt{2\,gh\left(\frac{W - W'}{W + W'}\right)} \qquad (60)$$

Inserting values W, W' and h = (5 and 15) feet, we have v = 1.05 f. p. s. (Living Room) and 1.8 f. p. s. (Chamber 2). Velocities for any other height of column and for other temperatures may be obtained in like manner. Reducing the 91 and 71 gallons to cubic inches and dividing by the velocity per hour in inches gives .46 sq. in. and .21 sq. in. respectively. Since pipe sizes are measured on the internal diameter these values are equivalent to pipes of ¾- and ½-inch respectively. *For the determination of pipe sizes, friction included, see Art. 197.* The application of friction equations to pipes of 4 inches or more in diameter is very satisfactory but for small pipes, such as are found in the average house heating plant, it is still the custom to use tables of sizes based upon what experience has shown to be good practice. Such tables may be found in the Appendix. From Table 34 we find the branches and risers to the two radiators under consideration to be 1¼- and 1-inch respectively.

In *steam systems* where the heating medium is a vapor and subject in a lesser degree to friction, the discrepancy between the theoretical and the practical sizes of a pipe is not as great as in hot water. Each pound of steam at 220° in condensing gives off about 970 B. t. u. To supply the heat

loss of the Living Room, 15267 B. t. u., requires 15.8 pounds of steam per hour = .26 pounds of steam per square foot of radiation. As a general statement *use one-fourth of a pound of steam per square foot of direct radiation per hour.* To check this statement, each square foot of steam radiation gives off 250 B. t. u. per hour and will condense 250 ÷ 970 = .258 pounds of steam.

The volume of the steam per pound at the usual steam heating pressure 17 to 18 pounds absolute is 23 cubic feet. Since the above radiator requires 15.8 pounds per hour there will be needed 23 × 15.8 = 363 cubic feet per hour. With the velocity of the steam in the pipe lines 15 feet per second (900 ft. per min. about one-seventh that allowed for power plant machinery. Taken as a fair approximation for small pipes) the area of the pipe will be 363 × 144 ÷ 54000 = .97 sq. in. = 1⅛-in. diameter. For two-pipe connections a 1-inch pipe would be considered good practice, but for one-pipe connections where the condensation is returned against the steam, a 1¼-inch pipe would be required.

See Tables 38, 39, 40 and 41, Appendix, for sizes and capacities of pipes carrying steam. For a discussion of steam pipe sizes by rational equation, including friction, see Art. 197.

107. Proportioning Pipe Sizes for a Heating System:— Begin at the farthest radiator and proceed toward the boiler as shown in the following tabulations.

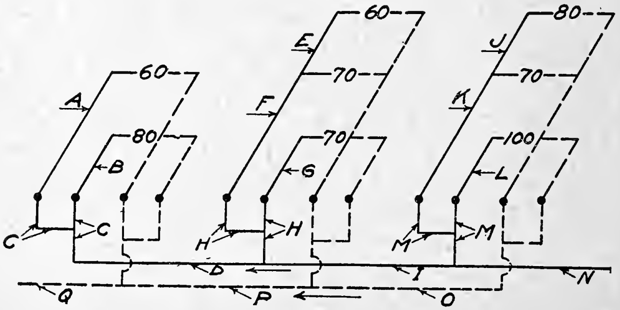


Fig. 99.

Basement Main Two-Pipe Hot Water System (Fig. 99).

TABLE XVII.

L. P. = Low Pressure Open Tank System. H. = Honeywell System.

Sq. ft.	Branch from Rad.		Riser		Branch from Main		Main	
	L. P.	H.	L. P.	H.	L. P.	H.	L. P.	H.
60	1	¾	A 1	A ¾				
80*	1½	1¼	B 1½	B 1¼	C 2	1½		
140							D 2½	D 2
60	1	¾	E 1	E ¾				
70	1	¾	F 1½	F 1				
70	1¼	1	G 1¼	G 1	H 1½	1½		
340							I 2½	I 2½
80	1	¾	J 1	J ¾				
70	1	¾	K 1¼	K 1				
100	1¼	1	L 1¼	L 1	M 2	2		
590							N 3	N 2½

* End of supply line, first floor radiator given advantage
Return branches same as supply branches
Return Main reversed.

O		P		Q	
L. P.	H.	L. P.	H.	L. P.	H.
2	2	2½	2½	3	2½

Basement Main Two-Pipe Steam System. Sealed returns. (See Fig. 99).

TABLE XVIII.

Sq. ft.	Branch from Rad.		Riser		Branch from Main		Main	
	S	R	S	R	S	R	S	R
60	1¼	1	A 1¼	1				
80	1½	1	B 1½	1	C 2	1½		
140							D 2½	Q 2
60	1¼	1	E 1¼	1				
70	1¼	1	F 1½	1¼				
70	1¼	1	G 1¼	1	H 2	1½		
340							I 3	P 2
80	1¼	1	J 1¼	1				
70	1¼	1	K 1½	1¼				
100	1½	1¼	L 1½	1¼	M 2	1½		
590							N 3	O 1½

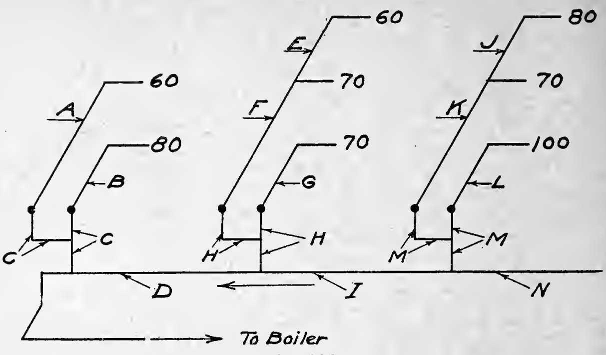


Fig. 100.

Basement Main One-Pipe Steam System (Fig. 100).

TABLE XIX.

Sq. ft.	Branch from Rad.	Riser	Branch from Main	Main
60	1½	A 1½		
80	1½	B 1½	C 2	
140				D 2½
60	1½	E 1½		
70	1½	F 2		
70	1½	G 1½	H 2	
340				I 3
80	1½	J 1½		
70	1½	K 2		
100	2	L 2	M 2½	
590				N 3

Return line to boiler same as dry return for two-pipe system, in this case 2-inch.

108. Pitch of Mains:—The pitch of the mains should be *not less than 1 inch in 10 feet for hot water systems and 1 inch in 30 feet for steam systems.* Greater pitches than these are desirable but are not always practicable. In hot water plants the one-pipe main has its highest elevation above the boiler and drops to the far end of the line, with the lowest point where it enters the boiler, the two-pipe basement main and return each pitch upward from the boiler to the end of the run and the attic main has its highest point at the top of the attic riser. The two pipe systems, both basement and attic mains, should have the supply and return reversed, i. e., the return

should begin at the first radiator served by the supply. In steam plants the supply main pitches downward toward the far end of each run, the highest point being above the boiler. The return main pitches downward from the end of each run toward the boiler.

109. Location and Connection of Radiators:—In locating radiators, it is usual to place them along the outside or exposed walls and when allowable, under the windows. This is probably the best location although some difference of opinion has been expressed on this point. When so placed the cold current of air from the window interferes with the warm upward current from the radiator and breaks it up. A series of tests reported in the Journal of the A. S. H. and V. E., July 1916, page 65, by a special committee of the society, shows that next to the center of the room the floor line near the outside wall is the most effective location. In buildings of several stories, the radiators should be arranged as far as possible in tiers, one vertically above another, thus reducing the number of risers and offsets. In one-pipe and two-pipe steam systems any number of radiators may be connected to the same riser, providing the riser is proportioned to the radiation supplied. In the two-pipe systems a water seal between each radiator and the return riser is advisable. This insures each radiator to be an independent unit in its action. In two-pipe hot water systems several radiators may have common flow and return risers as in steam systems providing the risers are carefully proportioned to the radiation. This is not always a safe plan. Under such an arrangement the upper radiators frequently have the advantage and rob the lower radiators. To be sure upon this point, either one of two methods may be employed: (1) offset the riser (See radiators *C* and *D*, Fig. 50); (2) isolate the returns (See radiators *A* and *B*, Fig. 49).

The connections from the risers to the radiators should be slightly pitched for drainage. They may be run along the ceiling below the radiator or above the floor behind the radiator. Connections should be at least 2 feet long to give flexibility for the expansion and contraction of the riser. For sizes of radiator connections see Table 41, Appendix.

110. General Application:—Figs. 101-104 show an illustrative layout of a hot water plant (See residence Art. 62). Because of the similarity between hot water and steam in-

stallations, the former only will be outlined. In making the layout of such a system, first locate the radiators in the rooms. This should be done with the advice of the owner who may have particular uses for certain spaces from which radiators must be excluded. Calculate the heat loss for each room, including exposure losses, ventilation losses, etc., and tabulate the results (See first column Table XX. Taken from Table XII). Calculate the square feet of radiation (Equation 51) and select the type, height and number of sections of each radiator from Table XIII. Check the radiator lengths and determine whether or not a radiator of such length will fit into the chosen space. If this can not be done, a radiator of greater height or number of columns must be selected. Branches from main to riser and riser sizes are usually the same although on a long branch it may be found necessary to put in a branch one size larger than the riser. Also, the branch from the riser to the radiator and the radiator connection sizes are usually the same excepting where there may be unusual conditions to meet, in which case the branch may be made one size larger than the standard connection. For commercial sizes see Tables 38 and 40, Appendix. Column in Table XVII marked "Radiators installed" should read "number of sections, height in inches and number of columns" (See Living Room = 18-38-3).

Locate the risers on the second floor plan and transfer these locations to the first floor and basement plans. Treat the first floor risers in a similar manner. The basement plan will then show by small circles the location of all risers. This arrangement will aid greatly in the planning of the basement mains. The principal features in the layout of the basement mains should be simplicity and directness. If the riser positions show approximately an even distribution around the basement, it may be advisable to run the main as a complete circuit system. If the riser positions show aggregations at two or three localities, two or three mains running directly to these localities are the most desirable. As an illustration the basement plan shows three clusters of riser ends, one under the kitchen, another under the study, and a third along the west side of the house. This condition immediately suggests three supply mains. That toward the north supplies the bath, chamber 4 and the kitchen, a total of 161 square feet. Being approximately 13 ft. long, a 1½-

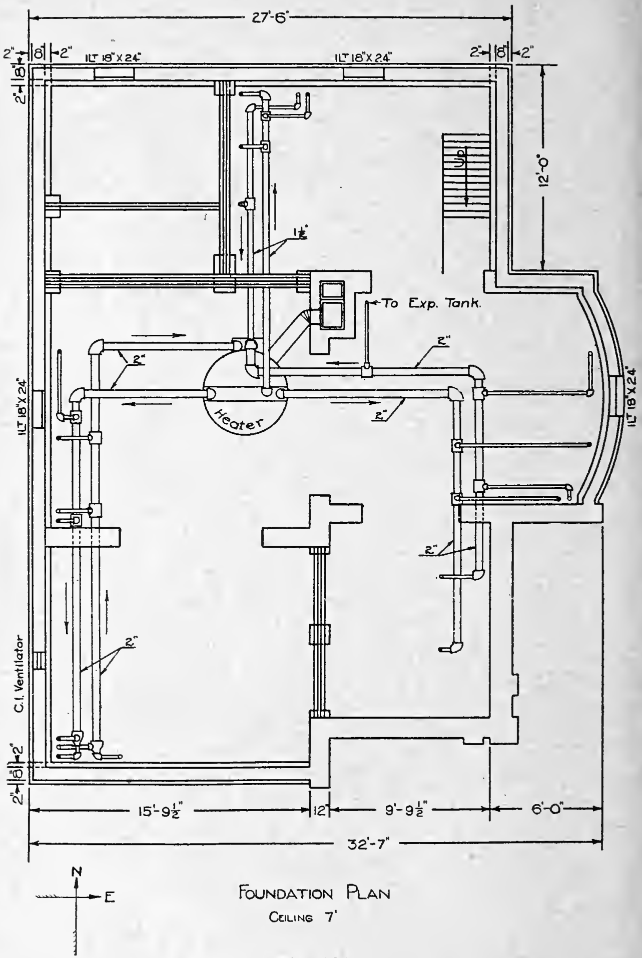
inch main will carry the radiation. That toward the east supplies chamber 1, the hall and the study, a total of 225 square feet, which can be carried by a 2-inch pipe. That toward the west side of the house supplies chamber 2, chamber 3, the living room and the dining room, a total of 277 square feet, which should have a 2-inch main.

In hot water work, as well as in steam, it is customary to connect supply branches (especially first floor branches) from the top of the mains, thus aiding the natural circulation. If this is not possible connect at 45 degrees. With a short nipple, a 45 degree elbow and a horizontal run of 2 to 3 feet this arrangement has sufficient flexibility to avoid expansion troubles. First floor radiators and those farthest from the boiler should be given the advantage of top branch connections and larger proportional pipe sizes.

In the selection of the boiler estimate the grate size from the total heat loss and check by the catalog rating in square feet of radiation. With soft coal at 13000 B. t. u. per pound, an efficiency of 60 per cent. and 5 pounds of coal per square foot of grate per hour there will be needed $(110574 \times 144) \div .60 \times 5 \times 13000 = 408$ sq. in. grate area. Adding 50 per cent. (Art. 100) for soft coal gives 612 sq. in. This agrees with Am. Rad. boiler W-19-6, 1250 sq. ft. rating. Checking *radiation equivalent* = 663 sq. ft. calculated radiator surface + 25% mains, risers, etc., = 829 sq. ft. total radiation + 50 per cent. = 1236 sq. ft.

TABLE XX.

	Heat loss, H from Table XII	Rad. Surface $R = .006\ H$	Radiators installed	Length of Rad. installed	Branches and Risers. Supply and Return	Rad. connection
Living R.	15267	91	18–38–3	47	1¼	1¼
Dining R.	9956	60	16–26–3	40	1	1
Study	12948	78	19–14–F	57	1¼	1¼
Kitchen	12828	77	16–38–3	40	1¼	1¼
Reception H.	14059	84	17–38–3	42	1¼	1¼
Chamber 1	10583	63	17–26–3	42	1	1
Chamber 2	11770	71	19–26–3	57	1	1
Chamber 3	9092	55	15–26–3	38	1	1
Chamber 4	8892	53	15–26–3	38	1	1
Bath	5179	31	9–26–3	23	¾	¾
		663				



FOUNDATION PLAN
CEILING 7'

Fig. 101.

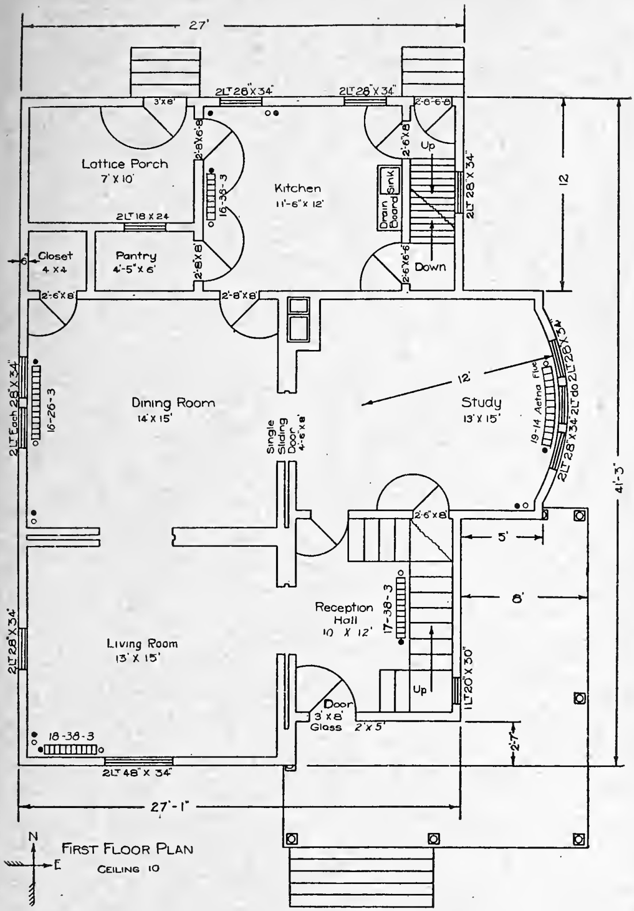
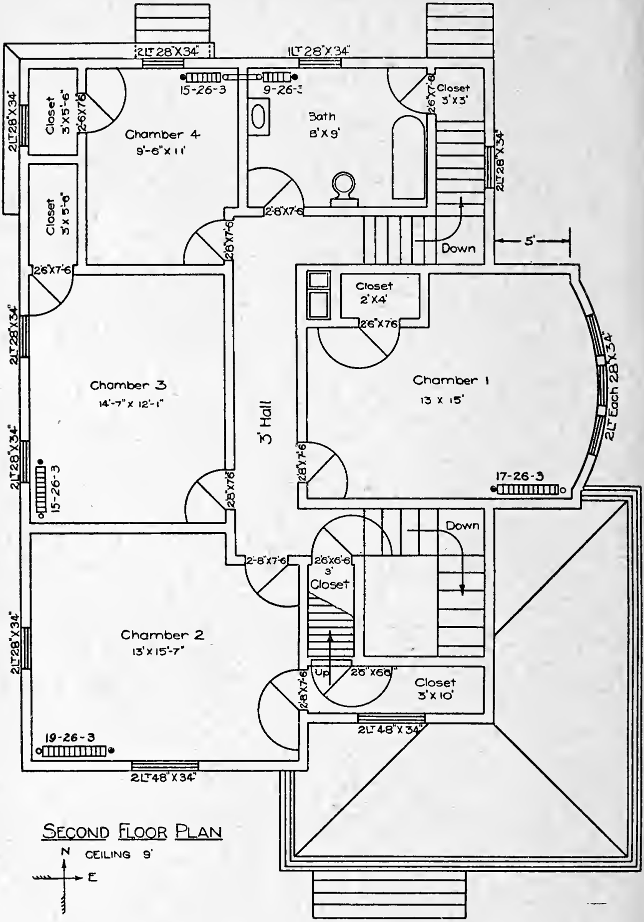

27'
3'x8'
2LT 28"X 34"
2LT 28"X 34"
2'-8"-6'-8
Lattice Porch
7' X 10'
Kitchen
11'-6" X 12'
16-38-3
2'-8"X6'-8
2'-6"X8'
Up
Sink
Drain Board
2LT 28"X 34"
12
2LT 18 X 24
Closet
4 X 4
Pantry
4'-5" X 6'
2'-8"X8'
2'-6"X6'-6
Down
6"
2'-6"X8'
2'-8"X8'
12'
2LT Each 28"X 34"
16-26-3
Dining Room
14' X 15'
Single Sliding Door 4'-6"X8'
Study
13' X 15'
19-14 Aetna Flue
2LT 28"X 34"
2LT do
2LT 28"X 34"
41'-3"
2'-6"X8'
5'
8'
Reception Hall
10' X 12'
17-38-3
2LT 28"X 34"
Living Room
13' X 15'
Up
1LT 20"X 30"
Door 3'x8'
Glass 2'x 5'
18-38-3
2'-7"
2LT 48"X 34"
27'-1"
N
E
FIRST FLOOR PLAN
CEILING 10

Fig. 102.

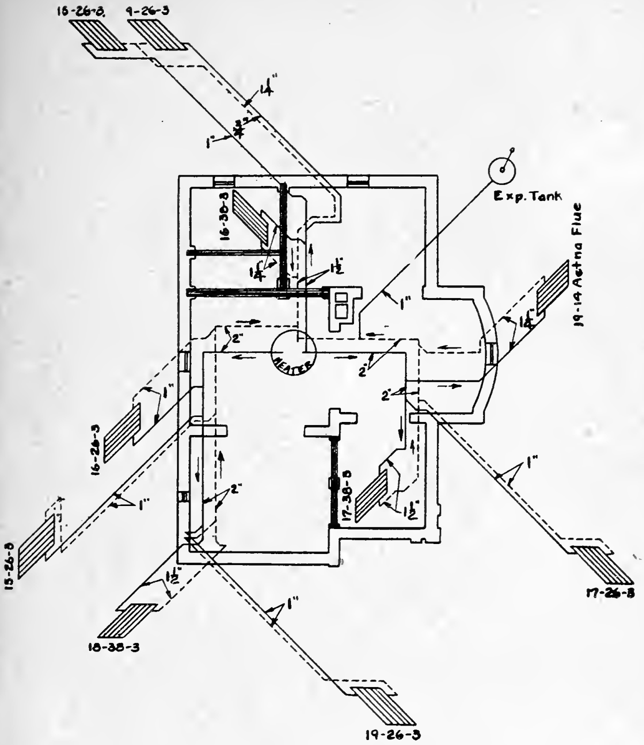


SECOND FLOOR PLAN

N CEILING 9'

E

Fig. 103.



MAIN AND RISER LAYOUT.

Fig. 104.

111. Insulating Steam Pipes:—In all heating systems, pipes carrying steam or water should be insulated to reduce the heat losses unless these pipes are to serve as radiating surfaces. In most plants the heat lost through these unprotected surfaces, if saved, would soon pay for first-class insulation. The heat transmitted to ordinarily still air through one square foot of the average horizontal wrought iron pipe is as great as 2.65 B. t. u. per hour, per degree difference of temperature between the inside and the outside of the pipe.

Assuming this to be 2.5, with steam at 100 pounds gage and air at 80°, the heat loss is (338 — 80) $\times$ 2.5 = 645 B. t. u. per hour. With steam at 50, 25 and 10 pounds gage respectively this will be 545,467 and 397 B. t. u. If the pipes were located in an atmosphere having decided air currents this loss would be much greater. The average unprotected low pressure steam pipe will probably lose between 350 and 400 B. t. u. per square foot per hour. Assuming this to be 375 and applying it to a 6-inch pipe 100 feet in length, for a period of 240 days at 20 hours a day, we have a heat loss of 171 $\times$ 375 $\times$ 240 $\times$ 20 = 307800000 B. t. u. With coal at 13000 B. t. u. per pound and a furnace efficiency of 60 per cent. this will be equivalent to 39461 pounds of coal, which at $3.50 per ton will amount to $69.05. From tests that have been run on the best grades of pipe insulation, it is shown that 80 to 85 per cent. of this heat loss may be saved. Taking the lower value there would be a financial saving of $55.24 if covering were used. If a good grade of pipe covering installed on the pipe is worth $85.00 the saving in one and one-half year's time would nearly pay for the covering.

To be effective, insulation should be cellular but should not permit air circulation. Small voids filled with still air are among the best insulators, consequently hair felt, mineral wool, eiderdown and other loosely woven materials are very efficient. Some insulating materials disintegrate after a time and lose their form but many patented coverings have good insulating qualities as well as permanency. Most patented coverings are 1 inch in thickness and may or may not fit closely to the pipe. A good arrangement is to select a covering one size larger than the pipe and set this off from the pipe by spacer rings. The air space between the pipe and the patented covering renders the covering more efficient. Table 50, Appendix, gives the results of a series of experiments on pipe covering, obtained at Cornell University under the direction of Professor Carpenter.

112. Water Hammer:—When steam is admitted to a pipe that is full of water, it is suddenly condensed causing a sharp cracking noise. The concussion produced may become so severe as to crack the fittings or open up the joints. The noise is due to the sudden rush of water from the surrounding space in an endeavor to fill the vacuum produced by the condensed steam. Steam at atmospheric pressure

occupies 1650 times the volume of the water that formed it, so when this steam is suddenly condensed a very high vacuum is produced which caused a relatively high velocity in the water adjacent to it. Steam should always be admitted very slowly to a cold pipe or to one filled with water.

Water hammer is frequently produced in water mains by suddenly stopping the stream of flowing water. For a theoretical discussion of this subject see Church's Hydraulic Motors, page 203. To find the approximate pressure p in pounds per square inch, produced by water hammer when v = velocity of the water in feet per second, use $p = 63\ v$. Also, to find the least time in seconds required in closing a valve on a water main that water hammer may be avoided, divide twice the length of the pipe stream by 4670 (See reference above). To illustrate. Water in a water main 500 feet long is flowing at the rate of 10 feet per second. If the water movement were suddenly stopped by closing the valve at the end of the main the pressure produced at the valve would be approximately 630 pounds per square inch. The least time of closing the valve to avoid water hammer would be $1000 \div 4670 = .21$ second.

113. Returning the Water of Condensation from a Low Pressure Steam Heating System to the Boiler:—In returning the water of condensation to a boiler four methods are in use; gravity, steam traps, steam loops and steam or electric pumps. The *gravity system* is the simplest and is used in all cases where the radiation is above the level of the boiler and where the boiler pressure is used in the mains. In a gravity return no special valves or fittings are necessary but a free path with the least amount of friction in it is provided between the radiators and some point on the boiler below the water line. No traps of any kind should be placed in this return circuit.

All radiation should be placed at least 18 inches above the water line of the boiler to insure that the water will not back up in the return line and flood the lower radiators. Flooding usually takes place through the return main and is the result of a restricted steam main. It may be due to a boiler which is too small and has to be forced thus causing siphonage. Where the radiation is below the water line, or where the pressure in the mains is less than that in the boiler, some form of *steam trap* or motor pump must be put

in with special provision for returning the water of condensation to the boiler. Two kinds of traps may be had, low pressure and high pressure. The first is well represented by the bucket trap, Fig. 105, and the second by the Bundy trap, Fig. 106. The action of these traps is as follows: Bucket trap.—Water enters at *D* and collects around the bucket

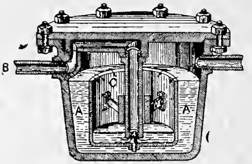


Fig. 105.

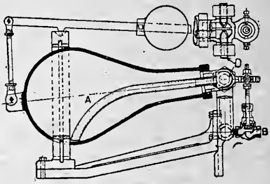


Fig. 106.

which is buoyed up against the valve. Water fills in and overflows the bucket until the combined weight of the water and bucket overbalances the buoyancy of the water. The bucket then drops and the steam pressure upon the inside, acting upon the surface of the water, forces it out through the valve and central stem to outlet *B*. When a certain amount of water has been ejected the bucket again rises and closes the valve. This action is continuous. Bundy trap.—Water enters at *D* through the central stem and collects in bowl *A* which is held in its upper position by a balanced weight. When the water collects in the bowl sufficiently to lift the weight, the bowl drops, valve *E* opens, and steam is admitted to the bowl thus forcing the water out through the curved pipe and valve *E*. This action is continuous.

Each trap is capable of lifting the water 2.4 feet above the trap for each pound of differential pressure. Thus, for a pressure of 5 pounds gage within the boiler and 2 pounds gage on the return, the water may be lifted 7 feet above the trap, or to the top of an ordinary boiler. This is not sufficient, however, to admit the water into the boiler against the pressure of the steam. A receiver should be placed here to catch the water from the *separating trap* and deliver it to a second trap above the boiler, which in turn feeds the boiler. Live steam is piped from the boiler to each trap, but the steam supply to the lower trap is throttled to give only enough pressure to lift the water into the receiver. A sys-

tem connected in this way is shown in Fig. 107. Here the receiver and trap are combined. Traps which receive the water of condensation for the purpose of feeding the boiler are called return traps and sometimes work under a higher pressure of steam than the separating traps. Many different kinds of traps are in general use but these will illustrate the principle of operation.

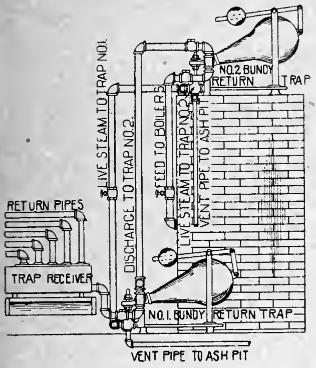


Fig. 107.

A very simple arrangement and a very difficult one to operate satisfactorily, is the *steam loop* (Fig. 108). The water of condensation from the radiators drains to receiver A, which is in direct communication with riser B. Drop leg D, being in communication with the boiler through a check valve which opens toward the boiler at the lowest point, is filled with water to point X sufficiently high above the water

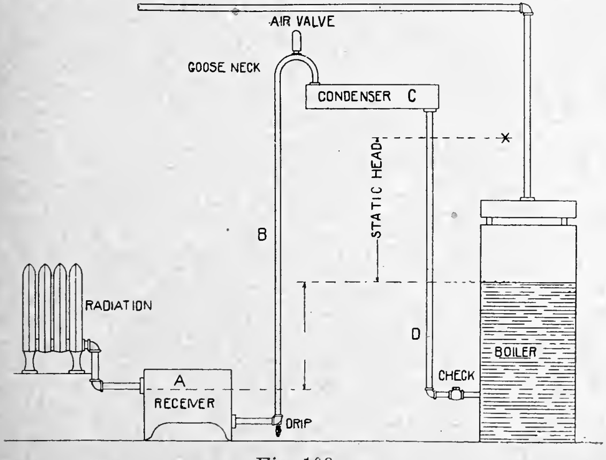


Fig. 108.

line of the boiler that the static head balances the differential pressure between the steam in the boiler and that in the condenser. Horizontal pipe *C* serves as a condenser which produces a partial vacuum and lifts the water from the receiver. This water is not raised as a solid body, but as slugs of water interspersed with quantities of steam and vapor. The water in *A* is at or near the boiling point and the reduced pressure in *B* re-evaporates a portion of it which, in rising as a vapor, assists in carrying the rest of the water over the gooseneck. When the condensation in *D* rises above point *X*, the static pressure overbalances the differential steam pressure, and water is fed to the boiler through the check.

To find the location of point *X* above the water line in the boiler, the following will illustrate. Let the pressure in the boiler, condenser and receiver be respectively 5, 2 and 4 pounds gage; then the differential pressure between the boiler and condenser is 3 pounds per square inch. If the weight of one cubic foot of water at 212° is 59.76 pounds, the pressure is .42 pound per square inch for each foot in height, i. e., one pound differential pressure will sustain 2.4 feet of water. With a pressure difference of 3 pounds this gives $3 \div .42 = 7.2$ feet from the water level in the boiler to point *X*, not taking into account the friction of the piping and check which would vary from 10 to 30 per cent. Assuming the friction to be 20 per cent. we have the static head $= 7.2 \div .80 = 9$ feet to produce motion of the water toward the boiler.

The length of riser pipe *B* and its diameter depends upon the differential pressure between the condenser and the receiver, and upon the rapidity of condensation in the horizontal. A differential pressure of 2 pounds will suspend $2 \times 2.4 = 4.8$ feet of solid water, but the specific gravity of the mixture in this pipe is much less than that of solid water. For the sake of argument let it be 20 per cent. of that of solid water, in which case we would have a possible lift, not including friction, of $5 \times 4.8 = 24$ feet. This is $24 - 9 = 15$ feet below the water level in the boiler. The diameter of the riser may vary for different plants, but for any given plant the range of diameters is very limited. These are found by experiment.

A drain cock should be placed in the receiver at the lowest point. When cold water has collected in the receiver it is necessary to drain this water to the sewer before the loop will work. An air valve should be placed at the top of the gooseneck to draw off the air. If the horizontal pipe is filled with air, there will be no condensation and the loop will refuse to work. Never connect a steam loop to a boiler in connection with a pump or any other boiler feeder. To determine whether a loop is working place the hand on the horizontal pipe. If this is cold it is not working.

The steam loop is used with success in factories and manufacturing plants in returning steam separator drips to boilers. In a series of experiments conducted in 1910, the condensation from four 100 square foot radiators was lifted 21 feet to a coil condenser and delivered by gravity to a

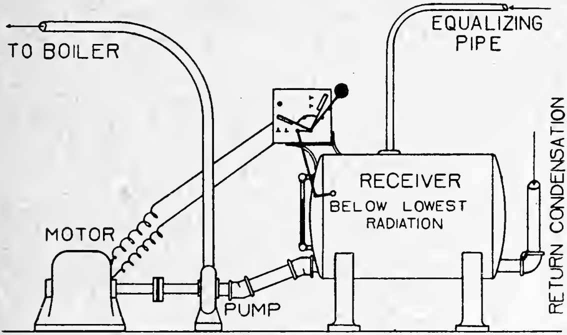


Fig. 109.

pressure tank located 5 feet above the receiver and maintained at a pressure 1½ pounds above the receiver pressure. Much experimentation will be necessary before the riser diameters and the condenser surfaces will be properly proportioned to be of general usefulness in heating systems.

The last method mentioned for feeding condensation to the boiler was a *steam* or *electric pump.* The operation of the steam pump is fully discussed in Art. 186. An electric motor-pump with its receiver and pipe connections is shown in Fig. 109. Its operation is very similar to that of the steam pump.

When the returning condensation fills the receiver to a certain level a float regulator starts the motor and pumps the water from the receiver to the boiler. When the water level drops the operation is reversed and the pump is automatically stopped. The motor pump is used on low pressure heating systems where the water of condensation from the coils and radiators drains below the boiler. If the boiler pressure were high the ordinary steam pump would be preferred. Where the pressure within the boiler is near that in the return main the operation of such a piece of apparatus is less expensive than that of the steam pump.

114. Hot Water Heating for Tanks and Pools:—The determination of the amount of heat transmitted, the amount of water heated and the square feet of coil surface needed for heating water by the use of immersed steam coils, follows closely the work given under hot water heating, Art. 101, Equations 49 and 50. From experiments conducted by the American Radiator Co., at the Institute of Thermal Research, the amount of heat transmitted in B. t. u. per hour, $K\ [t_s - (t_a + t_b) \div 2]$, through iron and brass pipes from steam (up to 10 lbs. gage) to water, allowing an efficiency of 50 per cent. for fouling of the pipes is:

Diff. between steam temp. and av. temp. of water, degrees	50	70	100	150	200
Brass	7200	12800	24000	48000	80000
Iron	4500	8000	15000	30000	50000

Knowing the amount of water to be heated through a given temperature difference, the coil surface and the steam condensed may be determined.

Application.—Required to heat 3000 pounds of water per hour from 60° to 90° with steam at 5 lbs. gage pressure. How many pounds of steam will be condensed per hour and how many square feet of iron coil surface will be necessary.

Result.—Steam temperature, 227°; average temperature of water, 75°; temperature difference, 152 degrees; heat given to water, 90000 B. t. u.; steam condensed, 92.2 lbs.; coil surface, 3 square feet.

115. Suggestions for Operating Hot Water Heaters and Steam Boilers:—Before firing up in the morning examine the pressure gage to see if the system is full of water. If there is any doubt, inspect the water level in the expansion tank. If it is a steam system, examine the gage glass and try the cocks to see if there is sufficient water in the boiler.

See that all valves on the water lines are open. On the steam system try the safety valve to make sure that it is free. Also see if the pressure gage stands at zero.

In starting a fire under a cold boiler it should not be forced but should warm up gradually.

For suggestions on firing read Art. 77.

In a boiler or heater using the same water continuously (the best plan) there will be little need of cleaning the inside of the boiler. Where fresh water is used frequently soft water should be used. Where hard water is used the boiler should be blown off and cleaned once or twice a month.

Never blow off a boiler while hot or under heavy pressure.

Inspect the pressure gage, glass gage, water cocks and thermometers frequently.

In case of high steam pressure in a steam system, cover the fire with wet ashes or coal and close all the drafts. Do not open the safety valve. Do not feed water to the boiler. Do not draw the fire. Keep the conditions such as to avoid any sudden shock. After the steam pressure has dropped sufficiently examine the safety valve.

Excessive pressure may be caused by the sticking of the safety valve in the steam system, or by the stoppage of the water line to the expansion tank in the hot water system. The safety valve should never be allowed to lime up, and the expansion tank should always be open to the heater and to the overflow.

In case of low water in a steam system, cool the fire, lower the pressure to atmosphere and fill the boiler.

When leaving the fire for the night shake down and bank as stated in Art. 77.

CHAPTER IX.

MECHANICAL VACUUM HEATING SYSTEMS.

116. Return Line Systems.—Air and Condensate Combined:—The term "vacuum heating" may properly be applied to that class of heating systems having a continuous negative pressure within the return main, the pressure within the radiators being controlled by the interposition of some form of thermal or float valve between the return main and the radiators. The vacuum may be produced by pumps or ejectors. In point of design this is the extreme in its departure from the low pressure gravity system and has the following advantages over it:

1. A positive and rapid return of the water of condensation.
2. In case of improper alignment of main and return pipes the negative effect of water and air pockets is reduced to a minimum.
3. Radiation at low levels may be drained by maintaining a vacuum in the return line proportional to the lift of the water of condensation.
4. Smaller return pipes may be used than are used on the ordinary gravity systems.
5. A continuous withdrawal of the entrained air from the radiators with the water of condensation. This insures a high efficiency of all the heating surface. This statement may not hold good for high radiators (36 to 48 inch) on the extreme end of a long heat run. On such radiators, air valves may be necessary.
6. This system is especially adapted to the use of exhaust steam with its extra large air and water content.
7. Comparative freedom from pounding and water hammer.

On the other hand there is an additional cost in maintaining the vacuum, and its use is restricted in small plants because of the extra cost of installation and superintendence.

Mechanical Vacuum systems of heating are frequently installed in connection with lighting or power units in which

case the exhaust steam may be used to supplement the live steam for heating. This substitution results in a great economy for the plant. A diagrammatic view showing the principal apparatus involved in such a plant is shown in Fig. 110. Live steam is connected to the power units and to the heating main, the latter through a pressure reducing valve to be used only when exhaust steam is insufficient.

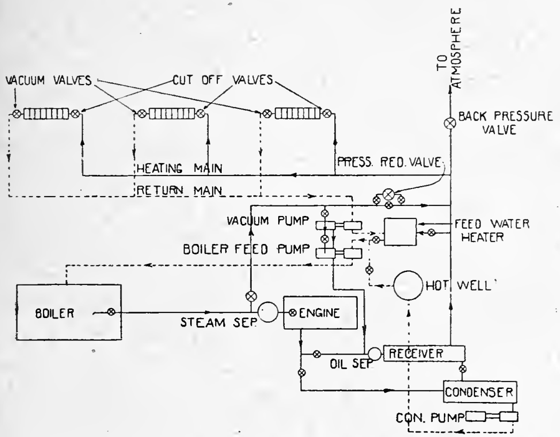


Fig. 110.

Exhaust steam from the power units is connected to the heating main and to the feed water heater. This exhaust steam line opens to the atmosphere through a back pressure valve which is set at the desired pressure for the supply steam. Oil separators remove the oil from the exhaust steam and deliver it to the oil traps. Boiler feed pumps and vacuum pumps, with the accompanying valves and governing appliances, complete the essentials of the power room equipment.

The steam supply to the heating system is piped to radiators and coils in the ordinary way, with or without temperature control. Thermostatic valves, or equivalent, are placed at the return end of each radiator and coil, and the returns from these are brought together in a common return which leads to the vacuum pump or ejector. The size of the return pipe and specialty valve for any one unit is

usually ½-inch, increasing in size as more radiating units are taken on. Horizontal steam mains usually terminate in a drop leg which is tapped to the return 8 to 15 inches above the bottom of the leg. Each rise in the system has a drop leg at the lower end of the rise. These points and all others where condensation and dirt may collect are drained through special separator valves to the return. Steam is carried in the main slightly above atmospheric pressure and just enough vacuum is maintained on the return to insure positive and noiseless circulation. In many cases where special lifts are required, these return systems are run under pressures 6 to 10 inches of mercury below atmospheric pressure. Under such conditions water may be lifted from 6 to 10 feet. Either closed or open feed water heaters may be used.

When water of condensation at 212° or above is released from the radiator through the vacuum valve to the returns with pressure below atmosphere, the result is a very

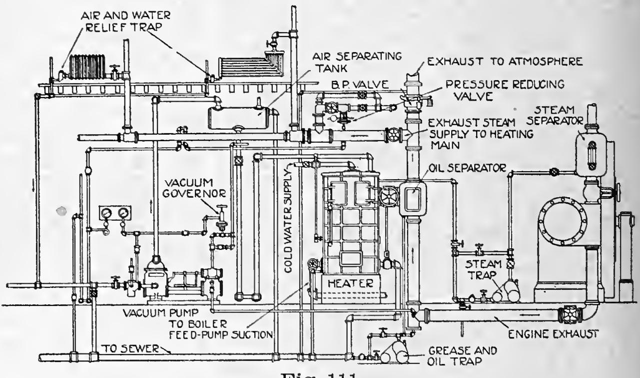


Fig. 111.

quick withdrawal and a partial re-evaporation of the water into steam or vapor at the lower pressure. If large amounts of condensation were thus released at one time, the vacuum would be temporarily broken, but being divided into a large number of small units having no regularity in the time of action, there is little difficulty. Although the vacuum is supposed to extend only from the pump to the vacuum valve, it may extend to within the radiator if the vacuum valve is set for a constant discharge. Such an arrangement cannot

be justified from the standpoint of economy, since the latent heat of all the leakage steam is lost from the heating system and thrown away. Just before entering the vacuum pump, the steam and water vapor mixed with the return water may be condensed by a spray of cold water. This spray assists in increasing the vacuum. From the vacuum pump the returns go to a feed water heater (open or closed) and from this by a boiler feed pump back to the boiler. The *Webster system*, shown in detail in Fig. 111, is a typical representative of this class. In all essential features this figure may stand for a number of others, among them the Dunham, the Bishop and Babcock, the Illinois and the Automatic. For comparative sizes of gravity and mechanical vacuum return pipes see Table 43, Appendix.

117. Air Line Systems.—Air and Condensate Separate:— Representing this type of heating is the so-called Paul system. It is usually installed as a one-pipe system, fed from overhead supply and drained to a wet return, although it may be connected up as a two-pipe system or fed from a basement supply. The air pump handles the water of condensation but is not installed as a vacuum producing agent. The vacuum in the air line connecting with the air valves at the radiators, is produced by a steam, air or hydraulic ejector which discharges directly into the atmosphere, into the atmospheric end of the exhaust heating main or into a secondary radiator where a separation is made, the water dropping to a receiver to be further used and the air exhausting to the atmosphere. This system differs from the ones mentioned in two essential points; first, the vacuum effect is applied at the air valve and the flow of water of condensation is independent of the vacuum; second, the vacuum effect is produced by the aspirator principle using water, steam or compressed air as motive power. The vacuum is supposed to extend only to the air valve at the radiator, but if desired this valve may be adjusted so that the vacuum may have an effect within the radiator. The layout of the system for large plants is about that shown in Fig. 112. This system is especially adapted to small plants having one-pipe complete circuit mains, because of its effectiveness in removing air from one-pipe radiators. When thus used the pump is omitted and the condensate flows direct to the boiler by the one-pipe gravity method.

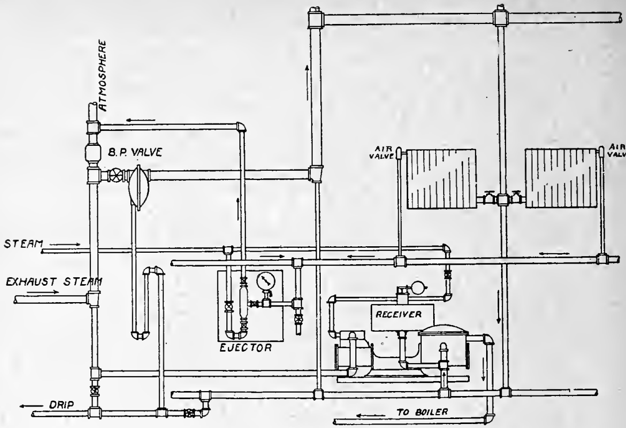


Fig. 112.

Fig. 113 shows typical vacuum connections between one-pipe and two-pipe radiators and the exhauster. Where electric

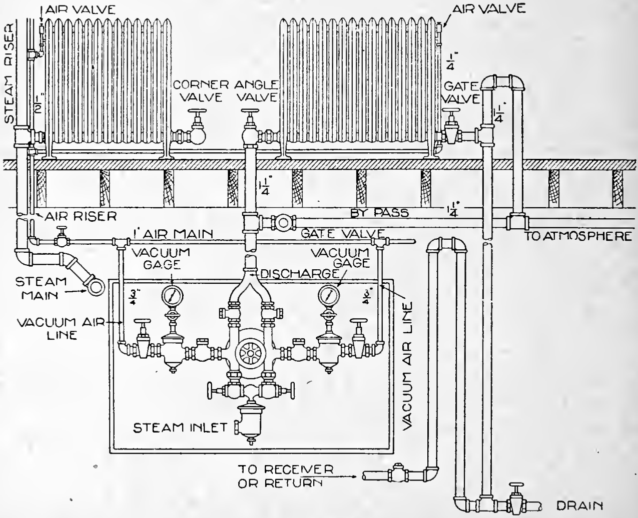


Fig. 113.

current is available exhausting may be done by the use of an electric motor pump.

118. Vacuum Pumps:—The satisfactory operation of vacuum heating systems depends upon the effective removal of air and water from the system. *Reciprocating pumps* (modified types of the direct acting piston pump) are generally used in producing the vacuum. Fig. 114 is a sectional view of the valve governing the action of the Ameri-

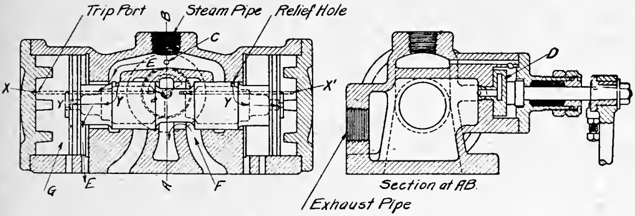


Fig. 114.

can-Marsh Vacuum Pump. Steam enters the chest through the pipe at *B* and a small amount passes through the port *C* to the auxiliary Valve *D*. Auxiliary Valve *D* is operated by a lever connected to the crosshead on the piston rod, and can be regulated by two adjusting screws. When the piston reaches the end of its stroke, steam is admitted by auxiliary valve through the small port *X* outside of the rear head of the main valve, forcing it forward to the position shown in the figure. When in this position steam travels through the steam port *E* and moves the piston to the opposite end of its stroke, the pump exhausting through the passage *F* as shown by arrows. Exhaust from the opposite end of the valve escapes through port *X'* and valve *D* to the main exhaust pipe. To hold the main valve in position after the auxiliary has placed it, live steam is admitted through balancing port *G* maintaining high pressure against the outside of the head, which more than balances the pressure on the inside of this same head owing to the difference of the area on either side. Port *Y* at each end of the valve prevents the valve from centering. In any position of the valve one of these ports is open to steam pressure and conducts steam to the outside of the valve head causing the valve to move into operating position. When the piston reaches the forward end of its stroke the operation is repeated at the for-

ward end of the valve. A few of the sizes and capacities of these pumps for the average mechanical vacuum heating system are given in Table XXI.

TABLE XXI.
Capacities of Marsh Vacuum Pumps.

Steam pressure 50 lbs. and above		Steam pressure 5 to 10 lbs. Not over 2 lbs. back pressure. For discharging into open receiver	
Size, inches	Sq. ft. direct rad.	Size, inches	Sq. ft. direct rad.
4x3x6	1200	7x3x8	1250
4x4x6	2200	8x3½x8	1800
4x5x6	3400	10x4x12	4000
4x5x8	4500	12x5x12	6500
5x6x10	8000	14x6x12	8500
5x7¼x10	12000	16x7¼x12	12000
6x8x12	18000	16x8x12	15000
8x10x12	30000	18x9x12	20000

Two systems of regulation are in common use in connection with piston vacuum pumps. In Fig. 115 the pressure in the return operates through the governor to regulate the supply of steam to the pump, thus controlling its speed. In Fig. 116 the pressure in the return controls the flow of injection

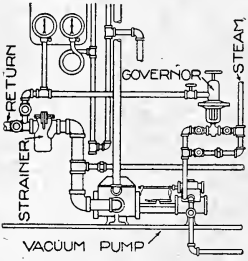


Fig. 115

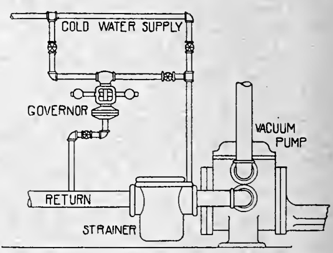


Fig. 116

water into the suction strainer and hence the rapidity of vapor condensation in the return. Either system provides automatic control for the vacuum. Injection water for the production of vacuum is not a necessity in vacuum returns. Systems are operating satisfactorily without it.

Occasionally it is desirable to have the *returns* for certain parts of heating systems *under different vacuum*. As an illus-

tration of this, suppose the returns for the radiators within a building are expected to carry condensate at atmospheric pressure and the returns from a set of heating coils in the basement condensate at four pounds below atmosphere. This may be accomplished by placing a pressure regulating valve in the branch requiring the least negative pressure (the higher line), as shown by Type *D* connection in the Web-

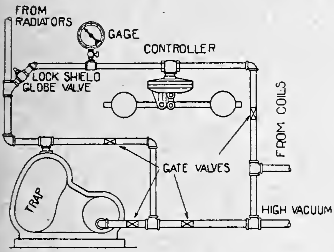


Fig. 117.

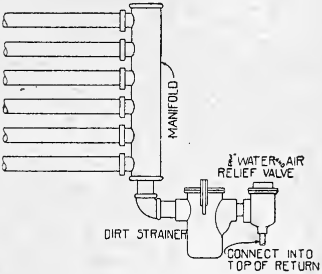


Fig. 118.

ester system, Fig. 117. The differential pressure between the atmospheric and vacuum lines may be varied to suit any condition by the controller valve. A trap and a controller valve are applied to each line having a different pressure from that in the main suction line.

Strainers or dirt catchers are installed next the pump on mechanical vacuum returns, to protect it from the cutting action of the core sand and dirt from the radiators. Where large amounts of radiation are grouped, a dirt catcher may be placed at the outlet of each group. (See Figs. 115, 116 and 118).

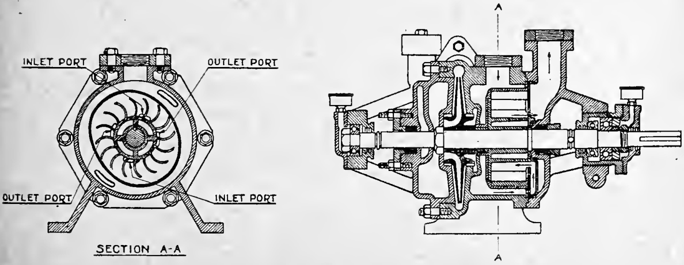


Fig. 119.

Centrifugal pumps are being increasingly used on vacuum returns where a moderate vacuum only is to be maintained. The Nash Hydroturbine (Fig. 119) represents this class. The pump consists of independent water and air units mounted on the same shaft. The water end is the usual centrifugal pump. The air and vapor pump (transverse section) shows a water wheel rotating in an elliptical casing partly filled with water. The water as it follows the wheel also follows the contour of the casing, this alternately moving from and toward the shaft and thus drawing in and exhausting continuously and without pulsation. In operation the centrifugal pump handles water in coming up to speed and continues automatic operation when there is a water supply. The air pump however produces continuous vacuum, but only at the rated speed. Table XXII gives recommended sizes and capacities of this pump.

TABLE XXII.

Capacities of Nash Vacuum Pumps.

Size	Sq. ft. direct equivalent radiation surface	Diameter orifice inches vac. 10 in.	Air capacity cubic ft. per min.	Water capacity gals. per min. 10 lbs. pres. 180° F.	Actual Horse Power	R. P. M.	H. P. of Motor
A	8000	9/64	6	11	.9	1800	1
B	16000	3/16	11	22	1.4	1800	1½
C	26000	1/4	19	35	2 0	1800	2
D	40000	9/32	25	60	2.8	1200	3
E	65000	3/8	42	90	3.9	1200	5

119. Vacuum Specialties:—Classified according to trade names these are:

Return Water Lines—radiator traps, thermo-traps, vacutraps, sylphon traps, radifiers and water seal motors.

Air Lines—vent valves, thermostatic valves, automatic expansion valves and vacustats.

Regardless of the trade names and the locations of the fittings on the systems, they may be classified under three heads:

Type A. Thermostatic valves—those opening and closing under the action of heat. Automatic and adjustable.

Type B. Float valves—those opening and closing under the action of the floatation or the impulse of the water of condensation. Automatic.

Type C. Orifice—those having a constant opening and leakage. Non-adjustable.

Thermostatic valves—Type A. Fig. 120 shows modifications of thermal control. The Webster composition expansion stem type, one of the earliest forms used on the mechanical vacuum systems is still used on many installations. The auto-

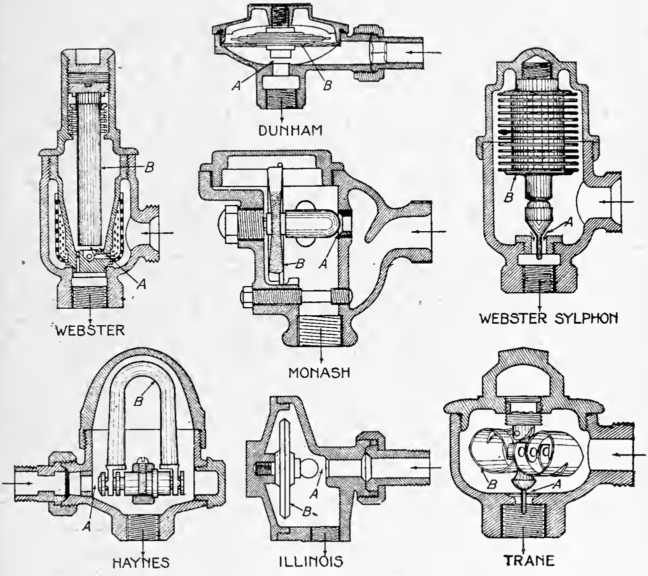


Fig. 120.

matic feature is the composition rubber stalk, which expands and contracts under change of temperature. The adjusting screw at the top permits the valve to be set for any conditions of temperature and pressure within the radiator. The water of condensation passes through a screen and comes in contact with the rubber stalk. The temperature of the water being less than that of steam the stalk contracts and the water is drawn through the opening *A* by the action of

the pump. As soon as the water has been removed steam flows around the stalk and expands it, closing the port. This process is continuous and automatically removes the

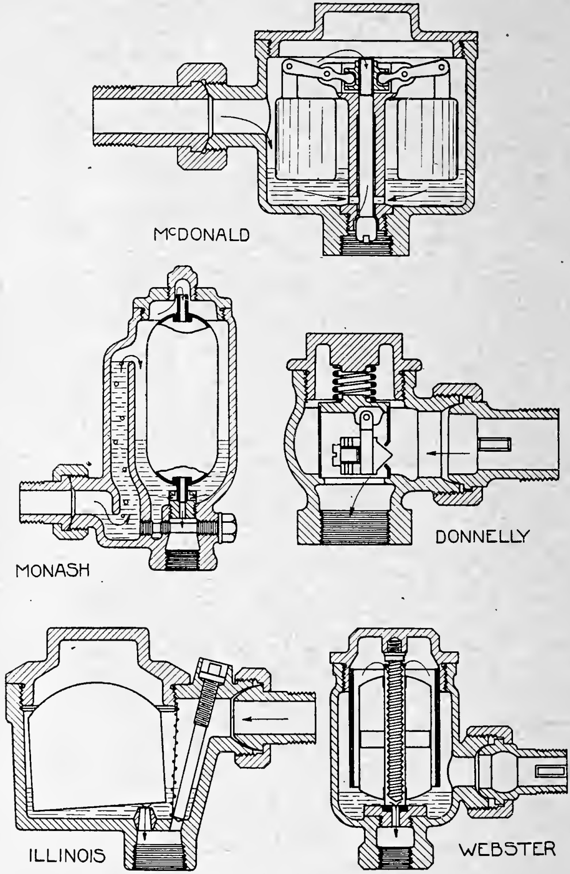


Fig. 121.

water from the radiator. The screen serves as a dirt catcher for the single unit. The other valves, excepting the Trane, have metal expansion chambers partially filled with liquids

that vaporize at temperatures between that of the steam and that of the returning condensation. In most cases the temperature approximates 200° F. The change in the vapor pressure within the enclosed chamber causes an expansion or contraction of the sides of the chamber thus closing or opening the valve. The expansion members of these valves differ in form and position. The Dunham, Monash and Illinois have single expansion chambers, the sylphon is multiple, or accordion form, and the Haines is a bent tube. The expansion member of the Trane valve is composed of two sheet metal coil springs which in turn are made of two thin pieces of dissimilar metals brazed together. Since the coefficients of expansion of the two metals are unequal, any change of heat coils or uncoils the springs thus opening or closing the valve. Other differences in these valves are to be found in the construction and location of the expansion member and in the style and location of the valve seat. The expansion member of four of the valves is in direct connection with the radiator steam. The rest are in connection with the return line. Two of the valves have flat seats, three have conical seats and two have line contact. Four seats are horizontal and three are vertical. Style *A* valves are automatic, positive, noiseless and adjustable. They may be used under either high or low differential pressures and may be used on either air or condensation lines.

Float valves—Type B.—When the differential pressure between the radiator and the return is very small and a fitting is desired that will serve merely as a separating trap to the radiator, a float valve or a valve actuated by the impulse of the water is frequently used. Fig. 121 gives five of the standard forms. There are five important features considered in the design of these float valves, continuous air removal, intermittent water removal, freedom from steam leakage, convenience in cleaning and freedom from noise. This is a combination that is difficult to obtain. The first three are points of efficiency and are not easily determined in the operation of the average plant except under test. So far there are few comparative data from which to draw conclusions. The fourth affects the attendant who has charge of the repair and upkeep of the plant, and the fifth is of vital interest to the occupant of the room. One of the objections frequently offered against the use of float valves

is the occasional noisy valve. When the differential pressure between the radiator and the return is so small that it is alternately changing positive and negative, there is liable to be a chattering of the valve, which is very annoying. This is not general but frequently obtains in one or more valves in a system.

Orifice—Type C.—In some systems the return fitting takes the form of a standard orifice which is non-adjustable and provides constant leakage. The use of such fittings is questionable.

Concerning the *economy of vacuum heating over low pressure heating,* many claims are made, some of which would be difficult to realize in practice. Estimates of saving range from 10 to 40 per cent. There are no doubt increased economies but these can not be stated in percentages. The two features of such a system that give decidedly increased efficiencies are the *use of exhaust steam and thermostatic control of the steam admitted to the radiator,* but each of these may be adapted to any system and consequently should not be credited here as economic features. On the other hand improvement in operating conditions is so marked that a general statement of higher efficiencies is justifiable. The claim is sometimes made that a mechanical vacuum system using exhaust steam as a heating medium may serve as a condenser to the engine and improve the efficiency of the engine to a marked degree. As a matter of fact this statement will seldom be justified. The back pressure on the engine will not drop below atmosphere except when the vacuum return valves are given a constant leakage, in which case there may be greater loss in the plant from the latent heat of the wasted steam than gain derived from the increased mean effective pressure in the engine. The one large economy to be looked for in heating systems lies in the use of exhaust steam as the heating medium. When we consider the fact that exhaust steam at atmospheric pressure contains 80 to 85 per cent. of the total heat of the live steam entering the cylinder (Art. 164), that this is all wasted when exhausted to the atmosphere, that the condensing engine saves only a small part of it and finally that the heating system may save it all, there is sufficient reason to look forward to its increased use in combined power and heating plants.

CHAPTER X.

MECHANICAL WARM AIR HEATING AND VENTILATION FAN-COIL SYSTEMS.

DESCRIPTION OF SYSTEMS AND APPARATUS EMPLOYED.

120. Elements of the Fan-Coil System:—In buildings having many occupants such as factories, school houses, theaters and auditoriums, a positive air supply to the rooms is usually required. To meet this condition there has been developed a type of heating and ventilating system variously known as the *fan-coil system*, *mechanical warm air system* or *plenum system*. This system contemplates the use of three distinctly vital elements: steam or hot water coils over which the forced air may pass and be heated, a blower or fan to propel the air and a proper system of ducts or passageways to distribute this heated air to the desired locations. Figs. 139 and 140 show these essentials in their relative positions. Attachments and improved mechanisms such as air washers and humidifiers, automatic air and heat control systems and brine cooling systems may be installed in connection with this type of heating but none of these auxiliaries change in any way the necessity for the three fundamentals named.

121. Variations in the Design of Fan-Coil Systems:—With regard to the position of the fan, two methods of installation are common. The first and most used is that shown in Fig. 122, where the fan is in the basement of the building and forces the air by pressure upward through the ducts and into the rooms. This arrangement gives the air within the building a pressure slightly greater than that of the atmosphere, causing any leakage to be outward from the rooms. A system so installed is a *plenum system*. The fan may, however, be placed in the attic (Fig. 123) with ducts leading to it from the rooms, in which case the air is pulled toward the fan thus causing the pressure within the building to be slightly less than that of the atmosphere. In the latter case the air is supposed to enter the basement inlet,

pass over the coil surfaces, and when heated pass by induction to the various rooms through the ducts. Since the leakage is inward, air from the outside readily enters at open windows and doors, breaking the vacuum effect of the fan and by-passing the heater, thus impairing the efficiency of the heating system. For this reason where heating is a vital factor, *exhaust systems without the aid of plenum systems are seldom installed.* Combined plenum and exhaust systems are to be recommended wherever the expense can be justified.

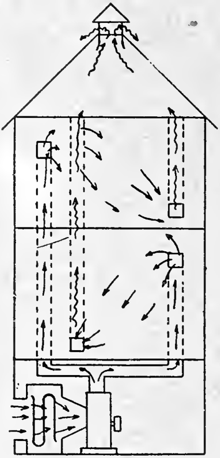


Fig. 122.

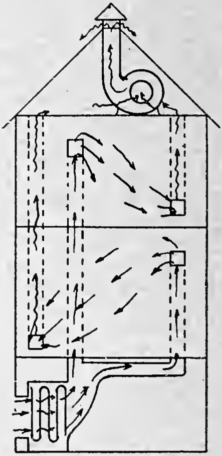


Fig. 123.

122. Blowers and Fans:—Many methods of moving air for ventilating and heating purposes have been devised. Some of these are positive at all times, others are dependent upon the existence of constant atmospheric air conditions and hence are a constant source of trouble. It is now a very generally accepted fact, that if air is to be delivered at definite times, in definite quantities and in definite places, it must be forced there by mechanical means. The recognition of this fact has led to a very common use of the mechanical fan or blower and its development to a fairly high efficiency.

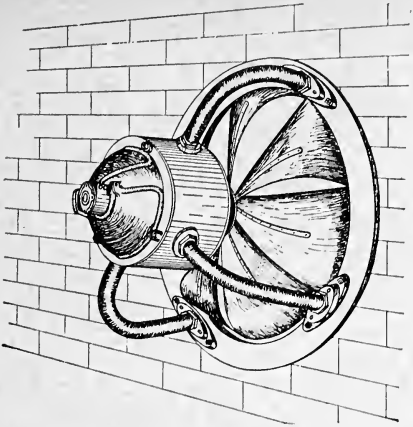


Fig. 124.

For exhaust service the fan generally used is of the *disk* or *propeller* blade type (Figs. 124 and 125). It is usually installed in the attic or near the top of the building, al-

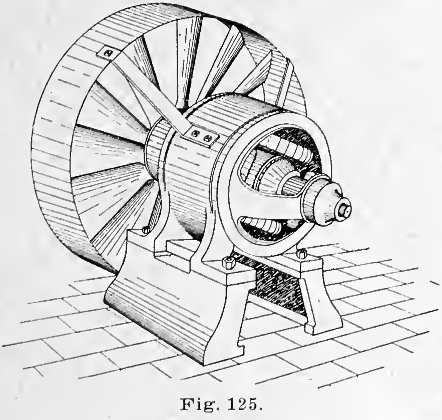


Fig. 125.

though in certain cases where the plenum fan is used for exhaust service it may be installed in the basement. The plenum system uses a centrifugal fan of the paddle wheel or the multiple blade type (Figs. 126 and 127). The first with

plane blades, called "steel plate" fan, is the old form of fan wheel and is now used on mechanical draft systems in power plants and on the cheaper plenum heating and ventilating plants. The second with curved blades, called "sirocco," "multivane" or "conoidal" fans by the respective companies, is a more recent development and is especially adapted to plenum plants. Tests of the multiple blade fans show higher efficiencies than are possible with the older forms. In the plenum systems fans are placed between the air intake and the heater coils or just following the heater coils (See Art. 125). For theoretical discussion of fans and blowers see Trans. A. S. H. & V. E., Vol. XXI, p. 43; Kent's M. E. Pocket-Book; Marks' M. E. Handbook; and Metal Worker, May 2, 1908, p. 44, serial.

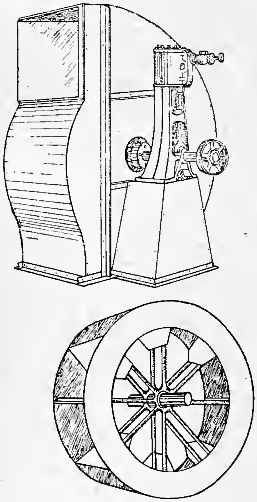


Fig. 126.

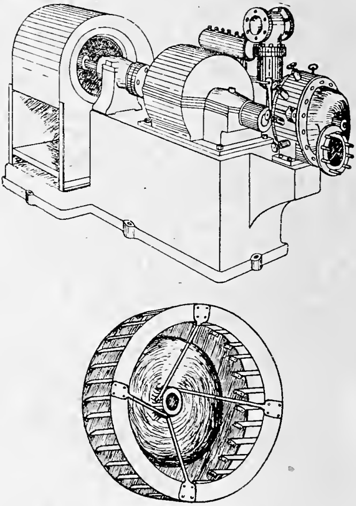


Fig. 127.

The motive power for fans may be of four kinds: electric motor or steam engine (or turbine), either direct connected or belted. Which one of these drives will be the most appropriate in any case will depend entirely upon local conditions and the nature of the available power supply. The steam driven engine or turbine unit is the most economical

since the exhaust steam from either may be used to supplement the live steam for heating (See Arts 164 and 172). The electric drive is the most convenient and in places where fans are employed for cooling and where steam is carried at pressures too low to operate the engines, motor drives should be installed. Electric motors are usually belted to the fans. This permits the installation of motors of smaller sizes and higher speeds at lower initial costs. Most of the larger engine driven units are direct-connected.

Fan housings are made in many different styles and of various materials, such as brick, wood, sheet steel or combinations of these. Steel housings are the most common and are made to fit any requirement. *Full housings* are those in which the entire fan wheel is encased and the entire unit is above the floor line. *Three-quarter housings* are those in which only the upper three-fourths of the fan wheel is encased, the completion of the air-sweep around the blades being obtained by properly forming the brick foundation upon which the fan is installed. Large fans are usually three-quarter housed, especially if they are to deliver air into underground ducts. Fig. 128 shows a three-quarter housing and Fig. 129 a full housing.

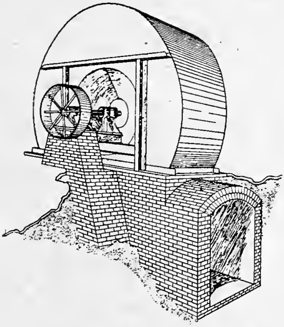


Fig. 128.

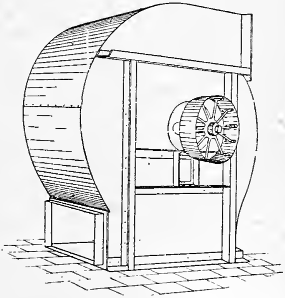


Fig. 129.

The circular opening in the housing around the shaft is the inlet to the fan, the air being thrown by centrifugal force to the periphery and at the same time given a circular motion, thus leaving the fan wheel tangentially through the discharge opening. Fans may be obtained which will deliver

air at any angle with the horizontal. They may have two or more discharge openings (multiple discharge fans as shown in Fig. 129), and double side inlets, i. e., air entering the fan from each side of the center. Where double side inlets are used they are smaller than the single side inlet for the same sized fan. All fan casements should be well riveted and braced with angles and tee irons. The shaft should be fitted with heavy pattern, adjustable self-oiling bearings, rigidly fastened to the casement and properly braced. The thickness of the steel used in the casement varies according to the size of the fan, from No. 14 to No. 11 for sizes in general use. The fan wheel should be well constructed upon a heavy spider to protect against distortion from sudden starting and stopping. Fans should be bolted to substantial foundations of brick or concrete. When connecting fans to metal ducts where sound from the fan may be transmitted to the rooms, the connection between the fan and the duct should be made through flexible rubber cloth.

The terms "Right Hand" and "Left Hand" refer to the position of the outlet relatively to a person facing the pulley or driving side of the fan, i. e., standing on the pulley or driving side of the fan, if the discharge is to your right, it is a right hand fan; if the discharge is to the left, it is a left hand fan.

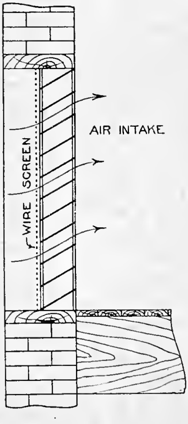


Fig. 130.

123. Fresh Air Entrance to Building and to Rooms:—Fresh air may enter through the building wall near the ground level or it may be taken from an elevation through a stack built for the purpose. In connection with washing systems it may be drawn from the nearest and most convenient source. Where no washing or cleaning systems are installed care should be exercised in selecting a location free from dust and other impurities. Where grills or shutters are placed in the opening, they are planned to obstruct the flow of the air as little as possible. Plain wire screens, ¾- to 1-inch mesh, should always be used to keep out leaves, birds and small animals. In exposed places stationary slats or grills should be put in and pitched to

keep out the rain. The amount of slope is dependent upon the width of the slat. In determining the net area of such a grill use the perpendicular distance between the slats and not the vertical spacing (See Fig. 130).

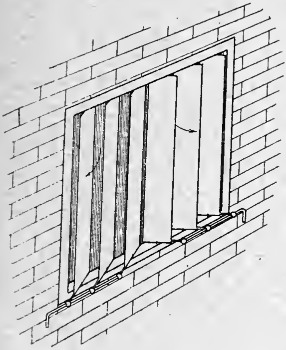


Fig. 131.

Air enters the rooms through registers, register faces or deflectors located above the heads of the occupants. Registers are used when volumetric regulation is not provided for elsewhere in the system. Register faces serve no economic purpose and are put in for ornamentation. Deflectors are frequently substituted for register faces to direct the air in definite lines about the room thus aiding uniform circulation. Where deflectors are used, registers and register faces are omitted. Fig. 131 shows the air inlet to a room with deflector attachment.

The construction of the dead end of the warm air stack is important. *Never finish to a square end* as in Fig. 132, *a*. Always have an easy curve as in *b*. This may be surfaced

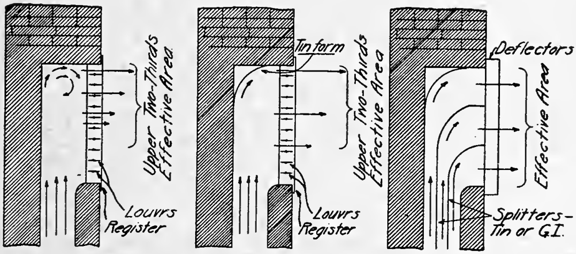


Fig. 132.

smooth with neat cement mortar over the rough bricks, but a better way is to have tin or light galvanized iron ends made to size and built in with the walls. These metal ends are still more efficient when fitted with *splitters* as shown in

c. It is found from tests that much more air is delivered through a given stack for the same power expenditure when these are used. In the average air inlet to a room the lower one-third of the opening is almost wholly ineffective. Where splitters are used each part of the opening is equally effective.

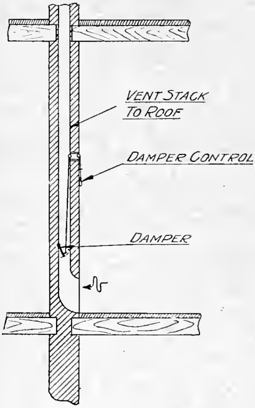


Fig. 133.

Vent openings are placed at the floor line and should be fitted with registers to close at night to avoid heat loss. Behind such registers there is always an accumulation of dust and dirt which is very unsanitary. When other means are provided for closing the vents, such as dampers within the stacks or at the top of the stacks in the attic, registers may be omitted and the vent openings curved and finished with cement flush with the floor line, as in Fig. 133. This permits the ducts to be more easily cleaned than where registers or faces are used. *Automatic air control from the fan room on all vents that lead to the attic is advisable.* In buildings having air supplied to two or more floors it is frequently necessary to condense the stacks into the smallest space possible. Fig. 134 shows a common arrangement. Notice that any vertical wall space may serve both as a heat stack for a lower room and a vent stack for an upper room. To accommodate large stacks the thickness of the wall is usually increased. All offsets must be made to fit the sizes of the standard bricks used.

Allowable velocities for net openings are higher than the corresponding velocities in furnace systems (See Table XXIV. Register sizes are given in Table 19, Appendix.

124. Heating Surfaces:—Heating surfaces used with plenum systems may be divided into two classes: *pipe coil surface* made of loops of 1- or $1\frac{1}{4}$-inch wrought iron or steel pipe and *cast surface,* made of hollow rectangular castings provided with numerous staggered projections to increase the outside surface and provide greater air and iron contact. To make a *heater* of either kind of surface, successive units are

assembled side by side, until the requisite total area and depth have been obtained. The total number of square feet of cast or pipe coil surface exposed to the air determines the

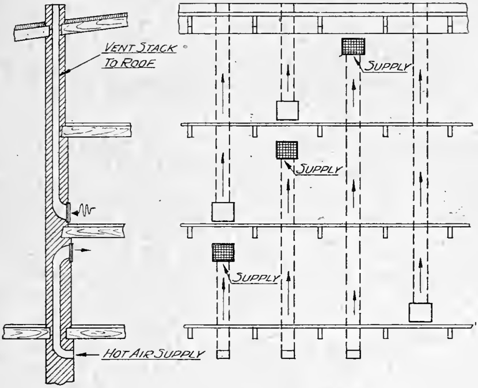


Fig. 134.

total number of heat units given to the air per hour, while the depth of the heater and the spacing of the coils control the final temperature of the air leaving the heater. Data upon these points have been obtained through exhaustive tests. Each must be considered in designing the heater system (See Arts. 136, 137, 139 and 140).

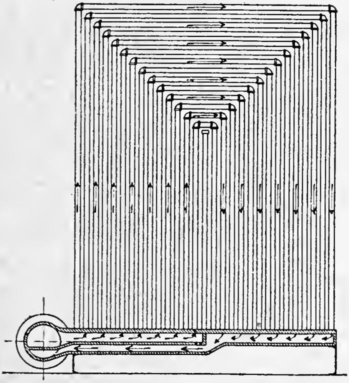


Fig. 135.

Pipe coils may be used with any steam pressure but cast coils should never be used with pressures exceeding 25 pounds per square inch gage. All plenum heating surfaces should be well vented and drained. Ample allowances also should be made for expansion and contraction.

Coil surface is of three kinds: that having the pipes inserted vertically into a horizontal cast iron header which forms the base of the section (Fig. 135); that having the pipes horizontal between two vertical side headers (Fig. 136); and that having one header vertical and one horizontal called the *miter coil* (Fig. 137). Sections fitted with pipe coils may be had two, three or four rows of pipes in depth. *The standard number of rows of pipes in any one section is four.* Sometimes these pipes are spaced in straight lines parallel with the wind and sometimes they are staggered. Staggered spacing increases somewhat the friction of the air current and the power of the fan. Heat efficiency tests of both straight and staggered spacings show little difference. Coil sections represented by Figs. 136 and 137 have better drainage than those shown by Fig. 135. In the latter the condensation must flow against the steam or be carried over with it to the return. All condensation that collects in the supply side of the header must drain to the return side through one or more small holes in the division plate. This method of drainage is not satisfactory because the total area of the openings is constant and the amount of condensation to be passed is varying, thus causing a clogging by extra condensation or a short circuiting of the steam to the return. The miter section in addition to perfect drainage, has perfect expansion, permitting every pipe to assume any position necessary to account for a reasonable change of length without causing breaking stresses in the pipe threads.

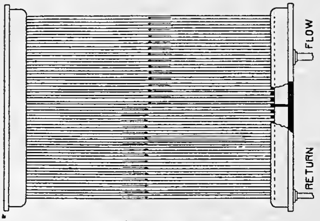


Fig. 136.

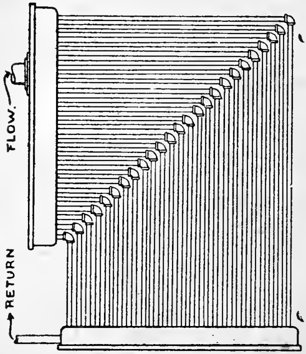


Fig. 137.

Cast iron radiating surfaces for plenum systems are cast in

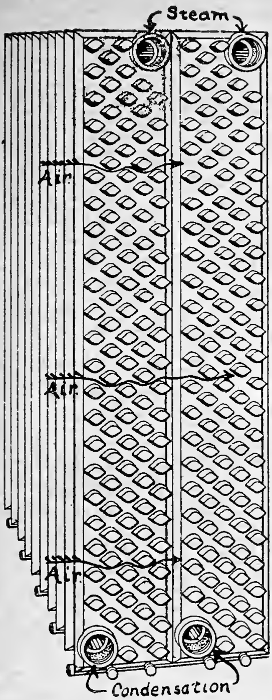


Fig. 138.

units called *sections*, and these are joined top and bottom by nipples into larger units called *stacks*, quite similar in all respects to a direct hot water radiator. Stacks are assembled one in front of another in the direction of the air current thus forming a *heater*. Fig. 138 shows a heater ten sections in width and two stacks in depth. Provided the conditions demand it, the heater may be built two or even three stacks in height, thus doubling or tripling the gross wind area (See Art. 140).

Cast iron heaters of the *vento* type are made in sizes shown by Table XXIII. It is unusual to assemble less than five or more than twenty-five sections to the stack. By the proper adjustment of number of sections to the stack and of stacks to the heater, any requirement of plenum system may be met.

Heaters are placed on either the suction or force side of the fan, usually the former in drying or evaporating plants and the latter in heating plants. Because of their weight, ample and firm foundations must be provided, with metal surface on top of foundation to permit expansion movement. In most installations for heating purposes, where both tempered and heated air are supplied, the heater is raised above the floor 18 to 30 inches to permit an air passage and damper for tempered air.

125. Division and Location of Coil Surface:—It is common practice to install heaters for plenum systems in two parts, known as *tempering coils* and *heating coils*. The total heating surface is first calculated and then divided into tempering and heating coils in desired proportions. The tempering coils are placed in the air passage just inside the in-

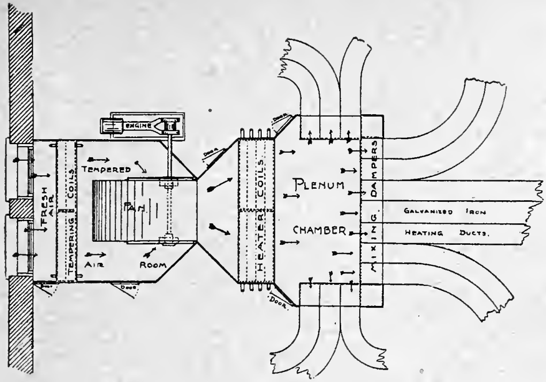
TABLE XXIII.

Vento Cast-Iron Heaters—Steam or Water.

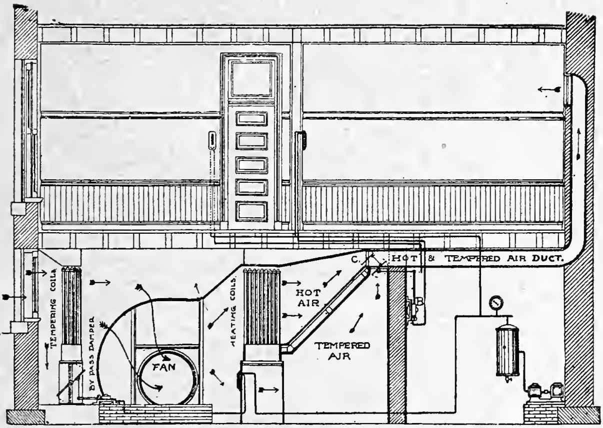
Narrow Sections	Sq. Ft. per Section	Height	Width
40 inch	7.50	$41\frac{1}{64}$	$6\frac{3}{4}$
50 inch	9.50	$50\frac{29}{32}$	$6\frac{3}{4}$
60 inch	11.00	$60\frac{11}{16}$	$6\frac{3}{4}$
Regular Sections			
30 inch	8.00	30	$9\frac{1}{8}$
40 inch	10.75	$41\frac{1}{64}$	$9\frac{1}{8}$
50 inch	13.50	$50\frac{29}{32}$	$9\frac{1}{8}$
60 inch	16.00	$60\frac{11}{16}$	$9\frac{1}{8}$
72 inch	19.00	72	$9\frac{1}{8}$

take of the building and usually contain from one-fourth to one-third of the total heating surface. In this way extremely cold air is tempered before it reaches the fan thus insuring good lubrication and preventing an accumulation of frost on the fan blades, which would seriously interfere with the free movement of the air. The heating coils are placed just beyond the fan on the force side (See Figs. 139 and 140).

Combined plenum and gravity-indirect systems (Fig. 141) have been installed in which the heating coils (tempering coils placed as before) have been divided and used in sections at the base of the stacks leading to the various rooms. Such an arrangement does not impair the plenum system and has an advantage in being able to by-pass the air through the plenum chamber and use the sectional heaters as an indirect-gravity system during the night and at other times when the fans are not running. With the coils divided as stated and the same amount of surface put in the indirect-gravity coils as would be required in the plenum heating coils, the gravity system will temper the room air but will not keep the rooms at the same temperature as when operating with the fan, because of the reduced volume of the air moving and the corresponding drop in efficiency of the heating surface. This difficulty may be overcome by installing more indirect-



PLAN



ELEVATION

Fig. 139. Fan Room Layout with Single Ducts along Basement Ceiling and all Mixing Dampers at Plenum Chamber.

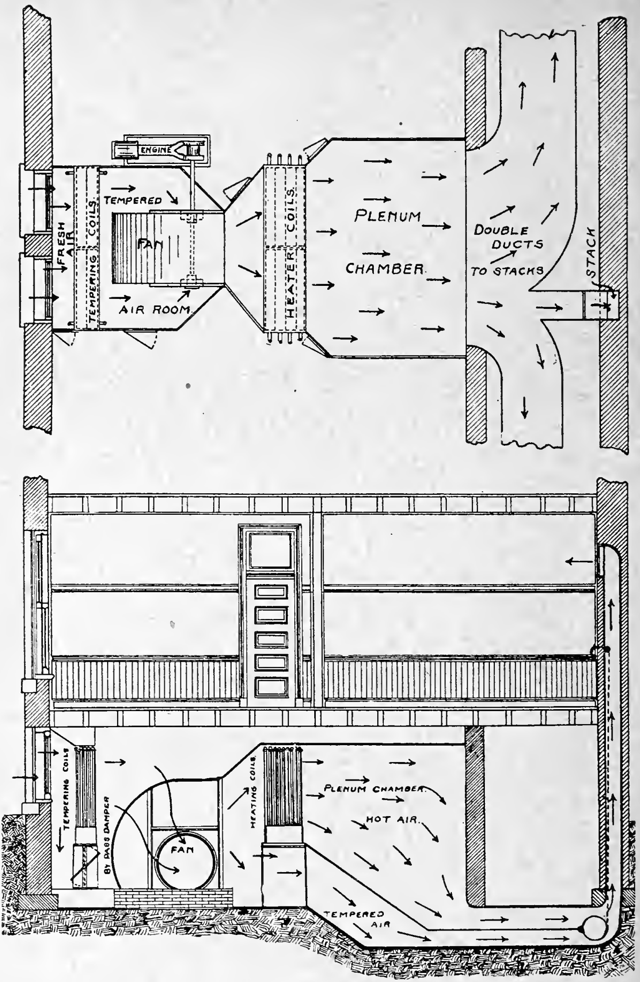


Fig. 140. Fan Room Layout with Double Underground Ducts and Mixing Dampers at Base of Room Stacks.

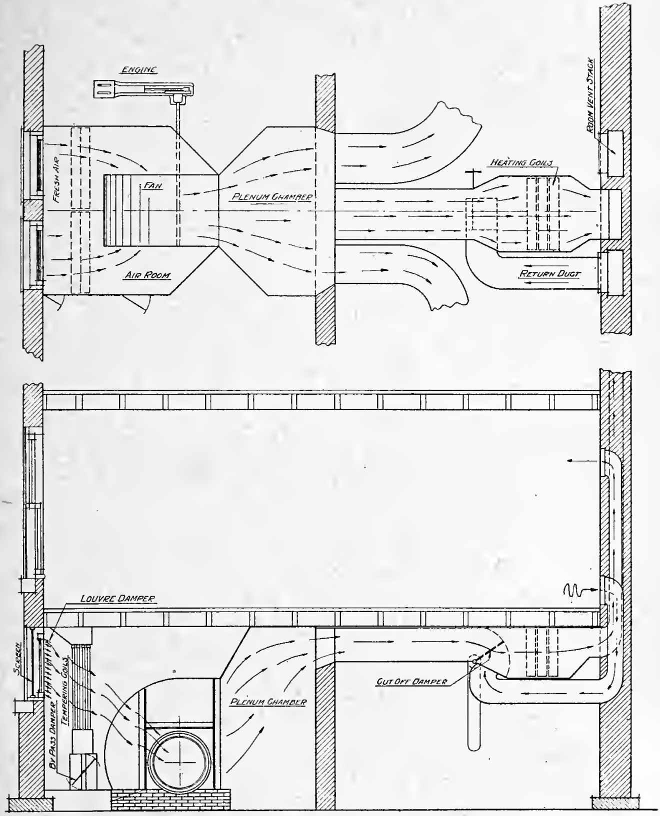


Fig. 141. Fan Room Layout with Heating Coils Divided into Individual Room Heaters.

gravity heating surface. In many plants (school buildings and the like) a moderate temperature (50° to 60°) throughout the night is all that is necessary and such an arrangement of coil surface is satisfactory (See Trans. A. S. H. & V. E., Vol. XIV, p. 96; Vol. XVII, p. 270; Vol. XVIII, p. 370).

In large installations where ventilation is of prime importance the ideal arrangement is the *split system*, i. e., plenum heating coils sufficient to heat the ventilating air from the outside temperature to say 80°, and direct radiation within the rooms sufficient to keep the room air at 60° during the night and at times when ventilation is not needed. This system is especially adapted to schools (Figs. 154 to 156) where for sixteen hours out of the twenty-four heating only is required. During the eight hours when air circulation is needed the amount of ventilating air may be regulated as desired, independent of the heating. In the split system automatic temperature control should be installed as a connecting link between the ventilating and heating systems. The temperature of the rooms on the coldest nights will be, say 60°. The radiators will be in service until with the aid of the plenum system (started at 8 to 8:30 A. M.) the room air is raised to 70° when the direct radiation is automatically thrown out of service. The radiators continue automatic action in connection with the plenum system holding the room temperatures within two degrees of fluctuation (generally 69° to 71°). When the plenum system shuts down all radiators throw on and night conditions prevail.

126. Single Duct Plenum System:—The duct systems that carry the air may be either of the single duct or double duct type. In both types of plants the fan delivers the air to a small room known as the *plenum chamber*. This chamber is divided into two parts, the upper one (hot air chamber) receives the air after leaving the heating coils; the lower one the air that has been warmed by the tempering coils. In the single duct system (Fig. 139) a single metal duct is carried from the base of each vertical heat stack to this plenum chamber and connected to both hot and tempered air through a mixing damper controlled by thermostat from the room supplied. Most ducts are carried along the basement ceiling and when the ceiling height is sufficient there is a false ceiling installed below the ducts for artistic effect. This system requires a complicated network of dam-

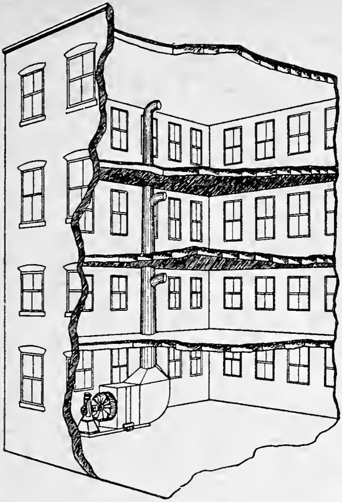


Fig. 142.

pers and ducts at the plenum chamber which to a certain degree limits its use. Fig. 142 shows a single duct installation applied to factories of several stories.

127. Double Duct Plenum System:—As the name indicates, this system (Fig. 140) runs all ducts in pairs (one above the other) from the plenum chamber to the base of each vertical room stack. The upper duct carries warm air from the heating coils while the lower duct carries tempered air. The mixing dampers are consequently located at the base of the vertical room stacks. The dampers may be hand controlled by chain pulls from the rooms above or automatically controlled by thermostats. With this arrangement it is evident that the principal ducts become *trunk lines* and are composed in a minimum of space.

Double duct systems are frequently installed as *subbasement systems*, as compared with the single duct systems which have usually metal ducts along the ceiling. Such ducts are below the basement floor and are constructed of brick and cement, or of concrete, about 4 inches thick. For designs of conduits, ducts and dampers see Figs. 139, 140, 151

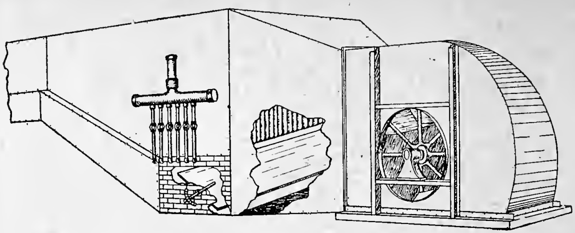


Fig. 143.

and 154. Fig. 143 shows a complete steel housed plenum unit of fan, coils, dampers and duct connections. For shapes and sizes of fans see manufacturers catalogs.

128. Air Washing and Humidifying Systems:—In connection with mechanical warm air heating and ventilating systems, there is often installed apparatus for washing and humidifying the air. In crowded city districts where the air is laden with dust, soot, the products of combustion and other harmful gases, its purification and moistening becomes a most important problem. The plenum system of heating and ventilating lends itself most readily to the solution of this problem, with the result that modern practice is tending more each day toward the combined installation of heating, ventilating and humidifying apparatus.

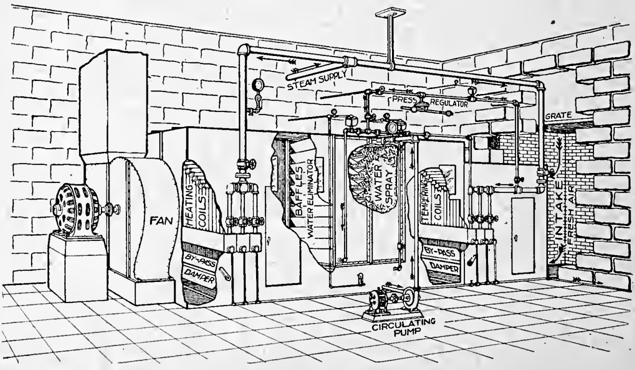


Fig. 144.

A purifier comprises two parts, a *washer* and an *eliminator* (See Fig. 144). The washer is located in the main air duct immediately behind the tempering coils, and provided with sheets or sprays of water through which the air must pass. Having caught the dust particles and dissolved some of the soluble gases from the air, the water falls to a collecting pan at the bottom of the spray chamber, and from there is again pumped through the spraying nozzles. As the water becomes too dirty or too warm, a fresh supply is delivered to the collecting pan. A small independent centrifugal pump is used for circulating the spray water.

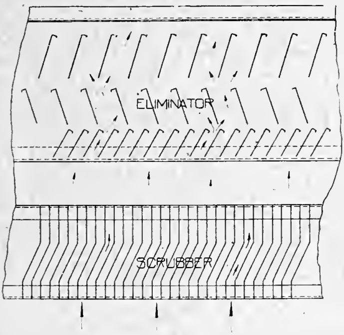


Fig. 145.

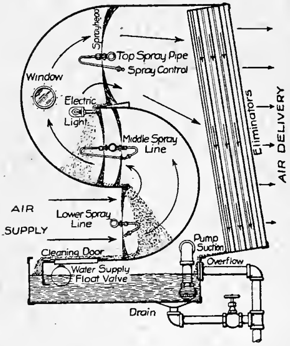


Fig. 146.

After passing through the washer, the air next encounters the eliminator, the purpose of which is to remove the surplus moisture, solids and water particles remaining suspended in the air. This is accomplished by baffle plates (Fig. 145), which change the direction of the air many times in succession and cause the water particles and solids to impinge upon the baffle plates and fall to the drip tank. As the air leaves the eliminator and enters the fan it may, with good apparatus, be relieved of 98 per cent. of all dust and dirt, may be supplied with moisture to very near the saturation point, and in summer time under favorable conditions, may be cooled from 5 to 15 degrees lower than the outside atmosphere. This is due to the cooling effect of vaporizing part of the water.

A purifier designed upon different lines than that in the last figure is shown in Fig. 146. In this the air makes a

double reverse curve and passes through four lines of spray before reaching the eliminator.

Fig. 147 represents the Zellweger combination fan and purifier which has proved very satisfactory in offices and small schools. This is similar to the ordinary blower fan excepting that the wheel is a combined filter ring and eliminator. Water may be circulated entirely or in part by the tan-

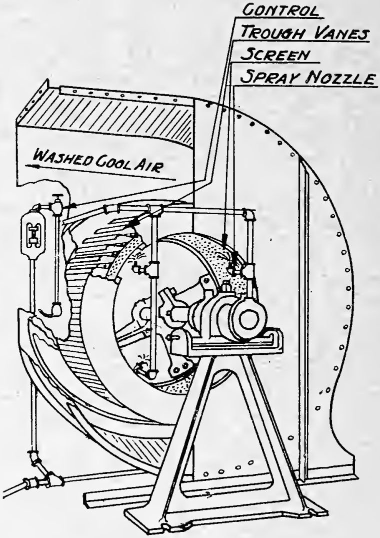


Fig. 147.

gential force of the water as it leaves the wheel. Air enters the wheel through the side opening and in passing through the wheel rim, which is composed of several layers of fine meshed wire cloth inside the curved vanes of the wheel, it comes in contact with the spray water from four spray heads. The outer edges of the blades are turned to form small gutters which catch the water and direct it to the large end of the wheel (wheel conical on surface) to the skimmer and collector ring for recirculation or for drainage. Full or three-quarter housing, single or double inlet and wheel diameters from 2.3 to 13 feet may be obtained.

An air washer well installed and maintained may be expected to remove 98 per cent. of all dust, dirt, soot, etc.; to lower the temperature of the air 85 per cent. of the initial wet bulb depression; to raise the temperature of the entering air from 35° to any temperature up to 60° and to add the necessary moisture to obtain any relative humidity up to 75 per cent. when the rooms are 70°; and to control the relative humidity within 5 per cent. variation when the wet bulb temperature of the air entering the purifier is below the desired dew point, all with a frictional resistance in the purifier not to exceed .2 inch water gage.

Special air cooling plants are installed in connection with the plenum system of ventilation, whereby refrigerated brine is circulated in the regular heating coils. (See Trans. A. S. H. & V. E., Vol. XV, p. 252).

CHAPTER XI.

MECHANICAL WARM AIR HEATING AND VENTILATION. FAN-COIL SYSTEMS.

AIR, HEATING SURFACE AND STEAM REQUIREMENT. PRINCIPLES OF DESIGN.

129. Definition of Terms:—Some of the technical abbreviations that are frequently used are the following: H = B. t. u. heat loss per hour by heat loss formula, H_v = B. t. u. heat loss per hour by ventilation, H' = total B. t. u. loss = $H + H_v$, Q = cubic feet of air used per hour as a heat carrier (calculated from H), $Q' = 1800\ N$ = cubic feet of air used in obtaining duct sizes, etc., when air for ventilation is in excess of air for heating, R = total square feet of heating surface in indirect heaters, t_s = temperature of the steam or water in the heaters, t = highest temperature of the air at the register (assumed the same as the temperature of the air upon leaving the heater), t' = temperature of the air in the room, t_v = temperature of the air at the register when Q' is used and t is reduced to keep the room from being overheated, t_o = temperature of the outside air, K = rate of transmission of heat through the coils, N = the number of persons to be provided with ventilation, V = velocity in feet per minute and v = velocity in feet per second. Other abbreviations are explained in the text.

130. Theoretical Considerations:—For illustrative purposes, references will be made throughout this discussion to a sample plenum design, Figs. 151, 152 and 153. These show the essential points of most plenum work and will serve as a basis for the applications. In any plenum design the following points should be theoretically considered for each room: heat loss, cubic feet of air per hour needed as a heat carrier (this should be checked for ventilation and the greater value used), net area of the register in square inches, catalog size of the register and area and size of the ducts. In addition to these the following should be investigated for the entire plant: size of the main leader at the plenum chamber, size of the principal leader branches, square

feet of heating surface in the coils, lineal feet of coils, arrangement of the coils in stacks and heaters, horse-power capacity and revolutions per minute of the fan including sizes of the inlet and the outlet of the fan, horse-power of the engine including the diameter and the length of stroke, and pounds of steam condensed per hour in the coils.

Fresh air is taken into the building at the assumed lowest temperature, $t_o°$, is carried over heated coils and raised to $t°$ (in certain cases to $t_r°$), is propelled by fans through ducts to the rooms and then exhausted through vent ducts to the outside air. It is the object of this section to so discuss this cycle that the principles may be applied to general problems.

131. Heat Loss and Cubic Feet of Air per Hour:—It is assumed in these calculations that the circulating air is all taken from the outside and thrown away after being used. Many installations have arrangements for returning part or all of the air to the coils and reheating it as an economic method of operation but this should not be taken into consideration in obtaining the sizes of the heaters. It is best to design the plant with the understanding that all the air is to be thrown away. It will then be large enough for any service that it may be expected to handle. Having found H by some acceptable equation (See Art. 39), the total heat loss is

$$H' = H + \frac{(Q \text{ or } Q')\ (t' - t_o)}{55} \text{ (See also Arts. 42 and 50).} \quad (61)$$

Assuming $t_o = 0°$ as the lowest temperature at which fresh air will be admitted (any temperature lower than this would call for recirculated air) this equation reduces to $H' = H + 1.27\ (Q \text{ or } Q')$. To determine whether Q or Q' will be used find Q from H, Equation 33, and compare with Q', taking the larger value. If this is to be a system of plenum heating only, let $t = 140°$, $t' = 70°$ and $t_o = 0°$, then

$$N = \frac{55\,H}{1800\ (t - t')} = \frac{H}{2290} = \text{approximately } \frac{H}{2300} \quad (62)$$

and since $t - t' = t' - t_o$, $H' = 2\,H$, that is to say, the heat given off from the air in dropping from the register temperature 140° to the room temperature 70°, goes to the radiation and leakage losses H, while that given off between the

inside temperature 70° and the outside temperature 0°, is charged up to ventilation losses H_v. Since these values are equal, $H' = H + H_v = 2\ H$.

Application.—Referring to Fig. 152, room 15, the calculated heat loss for this room, with $t' = 70°$ and $t_o = 0°$, is 70224 B. t. u. per hour; also, if $t = 140°$, $Q = 54775$ cubic feet per hour. Applying Equation 61, the total heat loss is 140448 B. t. u. per hour. With 54775 cubic feet of air sent to the room per hour, this provides good ventilation for thirty persons. Suppose, however, that fifty persons are to be provided for; this requires $50 \times 1800 = 90000$ cubic feet of air per hour. With this increased number of people in the room, the total heat loss is

$$H' = 70224 + \frac{90000\ (70 - 0)}{55} = 184864 \text{ B. t. u.}$$

Find H' when $N = 50$ and $t_o = -10°$.

132. Temperature of the Entering Air at the Register:—In plenum heating the registers are placed higher in the wall and the velocity of the air is greater than in furnace work. Suppose for this work the maximum temperature $t = 140°$ excepting where an extra amount of air is required for ventilation purposes, in which case the temperature must necessarily drop below 140° in order that the room will not be overheated, then

$$t = t' + \frac{55\ H}{Q'} \qquad (63)$$

Application 1. Referring to room 15 (compare with Art. 52) assuming the heat loss to have been estimated with ventilating air supplied sufficient for 50 persons, 90000 cubic feet per hour, the temperature of the air at the register is

$$t = 70 + \frac{55\ H}{90000} = 113°$$

Find t when $N = 40$ and $t_o = 20°$.

Application 2.—Referring to room 12 and assuming 200 persons in the room with 360000 cubic feet of air per hour, the temperature of the entering air will be 89°.

These applications do not take into account the heat given off by the audience, which would permit the reduction

of the entering air somewhat below the temperatures stated (See Art. 44).

The register air temperature is usually taken the same or slightly less than the temperature of the air upon leaving the coils. If this room were to be the only one heated, the coils would be figured for a final temperature of the air at 113°, but other rooms may have air entering at higher temperatures, consequently the temperature t upon leaving the coils should be that of the highest t at the registers.

133. Cubic Feet of Air Needed per Hour:—The amount of air needed per hour as a heat carrier (compare with Art. 50) is

$$Q = \frac{55\,H}{t - t'}; \text{ where } t = 140 \text{ and } t' = 70,\ Q = \frac{H}{1.27}$$

If extra air is needed for ventilation, $Q' = 1800\ N$.

134. Air Velocities, V, in the Plenum System:—Table XXIV gives the velocities in feet per minute that have been found to give good satisfaction in connection with blower systems.

TABLE XXIV.

Air Velocities in Plenum Systems.

	Fresh air intake	Over coils	Main duct near fan	Smaller branch ducts	Stacks	Reg'rs or other open'gs
Offices, schools, etc.	700 to 1000 F. P. M. Average 850 F. P. M.	800 to 1500 F. P. M. Av. 1000, deep heaters Av. 1200, shallow htrs.	1200 to 1800 say 1500	800 to 1200 say 900	500 to 700 say 600	300 to 400 say 300
Auditoriums, churches, etc.			1500 to 2000 say 1800	1000 to 1500 say 1200	600 to 800 say 700	400 to 600 say 400
Shops and factories			1500 to 3000 say 2000	1000 to 2000 say 1500	600 to 1000 say 800	400 to 800 say 500

135. Cross Sectional Area of Registers, Ducts, etc.:—With the above velocities in feet per minute, the square inches of net opening at any part of the circulating system may be obtained by direct substitution in the general equation

$$A = (Q \text{ or } Q') \times \frac{144}{60\,V} = 2.4\,\frac{(Q \text{ or } Q')}{V} \qquad (64)$$

The calculated main duct sizes refer to the warm air duct. The cold air duct in a double duct system need not be so large because on warm days when only tempered air is needed, the steam may be turned off from one or more of the heaters, permitting the warm air duct to furnish what otherwise would be required from the cold air duct. On account of this flexibility in the warm air system it seems necessary to make the cold air duct only one-half the cross sectional area of the warm air duct. For convenience of installation it would be well to make the former the same width as the latter and one-half as deep, unless by so doing the cold air duct becomes too shallow.

APPLICATION.—Assuming 2000000 cubic feet of air passing through the main heat duct at *A*, Fig. 151, per hour at the velocity of 1800 feet per minute, the duct is approximately 20 square feet in cross-section, or 2½ by 8 feet. The two main branches at *B* carry 800000 cubic feet per hour each at the same velocity and are 7.4 square feet in area, or 2 by 4 feet. The same branches of *C* carry 400000 cubic feet per hour each at a velocity of 1500 feet per minute and are 4.4 square feet in area, or 2 by 2½ feet, and branch *D* carries 300000 cubic feet at a velocity of 1200 feet per minute and is 1½ by 2⅔ feet.

The *stack sizes* are first calculated for a velocity of 600 feet per minute and then made to fit the laying of the brick work such that the velocities are 600 feet per minute or less. The net register is calculated for an air velocity of 300 feet per minute and the gross registers are taken 1 to 1.5 times the net area (the smaller value is used with splitters and diffusers).

136. Final Air Temperatures:—Since the amount of heat transmitted is directly proportional to the difference of temperature between the two sides of the metal, the first coils in a heater are the most efficient, this efficiency dropping off rapidly as the air becomes heated in passing over the coils. Final temperatures for different numbers of coil sections in banks have been found by experiment and may be taken from Table XXV (See also Table XXIX).

TABLE XXV.

Temperatures of Air upon Leaving Coils, Steam 227°, Air entering at 0°.

Sections	No. of rows	Velocities of air through coils in F P. M. 800	1000	1200	1500
1	4	42	33	28	23
2	8	71	62	56	52
3	12	96	87	80	75
4	16	119	108	101	93
5	20	136	125	116	108
6	24	153	140	131	120
7	28	169	155	143	131
8	32	183	166	154	141

These temperatures may be increased about 10 per cent. for 20 pounds gage pressure.

Table XXVI shows similar results quoted for the Vento cast iron heaters.

TABLE XXVI.

Temperature of Air upon Leaving Vento Coils, Steam 227°. Regular and Narrow Sections, 5 Inch Centers.

No. of stacks in depth		Velocities of air through coils in F. P. M. 800			1000			1200			1400		
	Ent. Temp.	0°	-10°	-20°	0°	-10°	-20°	0°	-10°	-20°	0°	-10°	-20°
1	Reg.	38			35			32					
	Nar.												
2	Reg.	68	62	55	62	56	49	58	51	44	54	47	40
	Nar.	51	43	36	46	38	31	43	35		40	32	
3	Reg.	93	87	82	86	80	75	81	75	69	76	70	64
	Nar.	70	64	58	65	58	52	61	54	47	57	50	43
4	Reg.	113	108	103	106	101	96	100	95	90	95	89	84
	Nar.	88	83	77	82	76	70	77	70	64	72	65	59
5	Reg.	129	126	122	122	118	114	115	111	107	109	105	100
	Nar.	103	98	93	96	91	86	90	84	79	85	79	74
6	Reg.	143	140	137	135	132	129	129	125	121	123	119	115
	Nar.	115	111	107	108	103	99	102	97	92	97	92	87
7	Reg.	154	152	150	147	144	141	140	137	134	135	131	128
	Nar.	127	124	120	120	116	112	114	109	105	108	103	99

137. Square Feet of Heating Surface, *R*, in the Coils:— To determine the number of square feet of heating surface in the coils of an indirect heater, the following equation may be used:

$$R = \frac{H'}{K\left(t_s - \frac{t + t_o}{2}\right)} \qquad (65)$$

Rule.—To find the square feet of coil surface in an indirect heater, divide the total heat loss from the building in B. t. u. per hour by the rate of transmission multiplied by the difference between the inside and the average outside temperatures of the coils.

Equation 65 presupposes a uniform rise in the temperature of the air as it passes over the coils, i. e., if the air is heated from 0° to 140° in passing over a heater 24 pipe rows deep, at the sixth row the temperature would be 35°, at the twelfth row 70°, and at the eighteenth row 105°. It is found that this is not the case but that the rise is more rapid in passing over the first part of the heater, gradually falling off to the end of the heater according to the logarithmic curve. The mean temperature difference between the inside and the outside of the heater instead of coming from an arithmetical mean as given within the brackets, Equation 65, is $\Theta_m = (\Theta_a - \Theta_b) \div \log_e\left(\frac{\Theta_a}{\Theta_b}\right)$ and the total number of square feet of radiation in the the heater is

$$R = \frac{H'}{K\left(\frac{\Theta_a - \Theta_b}{\log_e (\Theta_a/\Theta_b)}\right)} \qquad (66)$$

where Θ_m = mean temperature difference, Θ_a = temperature difference at the entering end and Θ_b = temperature difference at the leaving end. For full discussion of Equation 66 see Elements of Heat Power Engineering, Hirshfeld and Barnard, Chapter XXXV. Equations 65 and 66 give approximately the same results, as shown by Equations 67 and 68. The first is more easily applied and is recommended for ordinary heater calculations. These equations are rational and the terms indicated are readily apparent excepting perhaps the value *K*. Various experimenters have done extensive work toward establishing this and a few of their results will be briefly summarized.

Prof. Carpenter quotes extensively from experiments with coil heaters in blower systems and summarizes in the equation $K = 2 + 1.3\sqrt{v}$ where v is the average velocity of air over the coils in feet per second. With coil velocities in common use, 800 to 1400 feet per minute, this equation gives K from 7 to 8.5, which are very conservative and safe values.

Mr. F. R. Still gives the following equation for the *total* B. t. u. transmitted per square foot per hour, between the temperature of the steam and that of the entering air, *total B. t. u. transmitted* $= c\sqrt{v}\,(t_s - t_o)$, (Table XXVIII), in which v is velocity in feet per second and c is a constant which varies with the number of sections as shown in Table XXVII.

TABLE XXVII.

Values of c.

	Safe factor	Max. factor
1 section 4 rows of pipe	3.45	4.40
2 sections 8 rows of pipe	3.00	3.40
3 sections 12 rows of pipe	2.63	2.85
4 sections 16 rows of pipe	2.33	2.45
5 sections 20 rows of pipe	2.12	2.20
6 sections 24 rows of pipe	1.95	2.05
7 sections 28 rows of pipe	1.80	1.95
8 sections 32 rows of pipe	1.65	1.85
9 sections 36 rows of pipe	1.52	1.80
10 sections 40 rows of pipe	1.40	1.75

From the above values of c, Table XXVIII has been compiled, assuming $t_s = 227$, $t_o = 0$ and c = safe factor.

TABLE XXVIII.

Velocity of air in feet per min.	Total transmission in B. t. u. per sq. ft. per hour. $t_s = 227$; $t_o = 0$.							
	Rows of pipe deep							
	4	8	12	16	20	24	28	32
800	2840	2470	2164	1920	1750	1606	1450	1360
1000	3200	2790	2440	2170	1900	1810	1670	1535
1200	3500	3040	2670	2360	2150	1980	1825	1678
1400	3783	3290	2884	2555	2325	2138	1974	1809

Since the values given in Table XXVIII are the total B. t. u. transmitted per square foot per hour for different settings $= c\sqrt{v}\,(t_s - t_o) = K\left(t_s - \frac{t + t_o}{2}\right)$, K is found to vary between 10.5 and 15 with an average of approximately 12.

Table XXIX by Mr. C. L. Hubbard shows efficiencies that are less than those just considered. These agree more nearly with average practice.

TABLE XXIX.

Efficiencies of Forced Blast Pipe Heaters, and Temperatures of Air Delivered.

Velocity of air over coils at 800 feet per minute.

Rows of pipe deep	Temp. to which the air will be raised from zero			Efficiency of the heating surface in B. t. u. per sq. ft. per hr.		
	Steam pressure in heater			Steam pressure in heater		
	5 lb.	20 lb.	60 lb.	5 lb.	20 lb.	60 lb.
4	30	35	45	1600	1800	2000
6	50	55	65	1600	1800	2000
8	65	70	85	1500	1650	1850
10	80	90	105	1500	1650	1850
12	95	105	125	1500	1650	1850
14	105	120	140	1400	1500	1700
16	120	130	150	1400	1500	1700
18	130	140	160	1300	1400	1600
20	140	150	170	1300	1400	1600

For a velocity of 1000 feet per minute multiply the temperatures given in the table by 0.9 and the efficiencies by 1.1.

Perhaps the most extensive work along this line has been done by Messrs. Willis H. Carrier and F. L. Busey. For details see Trans. A. S. H. & V. E., Heat Transmission with Indirect Radiation Vol. XVIII, p. 172. From these experiments K varied from 9 to 13 for velocities from 800 to 1400 feet per minute.

When estimating plenum heating surface it is well to remember that after a coil has been in service for a time it

becomes somewhat less efficient than while clean and new, also that even though the very best arrangements for air removal are provided, these often fail or work with lessened efficiency. In general, even though transmission tests show rates of transmission as high as 12 or 13 it is much safer to take a lower value for average conditions. Assuming for illustration $K = 8.5$ and $V = 1000$ as the best values to use; also $t_s = 227$ (5 pounds gage pressure), $t = 140$ and $t_o = 0$, Equations 65 and 66 become

$$R = \frac{H'}{8.5\left(227 - \dfrac{140+0}{2}\right)} = \frac{H'}{1335} \qquad (67)$$

$$R = \frac{H'}{\dfrac{8.5\ (227 - 87)}{\log_e 227 / 87}} = \frac{H'}{1240} \qquad (68)$$

Note.—At the assumed rate of transmission, 8.5, each square foot of heating surface is equal to 5 square feet of direct radiation. This is due to the increased velocity of air over the radiating surface.

Cast iron heaters are being increasingly used for low pressure indirect heating, replacing pipe coil heaters. The efficiency of these heaters is, according to tests, about the same as that of the pipe coil heaters and hence Equations 65 and 66 will apply to both pipe and cast heaters. For more complete data on efficiencies and final temperatures than given in Table XXVI see American Radiator Co.'s Engineers' Data, Vento Heaters.

APPLICATION 1. *Where heating only is considered.*—Referring to Table XXXV, let H for the entire building ($t_o = 0°$) = 1483251 and $t = 140$. Then from Art. 133, $Q = 1156935$; by Equation 61, $H' = 2966502$; and Equation 67, the coil surface is

$$R = \frac{2966502}{8.5\left(227 - \dfrac{140+0}{2}\right)} = 2222 \text{ sq. ft.}$$

With three lineal feet of 1-inch pipe per square foot of surface we have 6666 lineal feet of coils in the heater.

APPLICATION 2. *Where heating and ventilation are combined.*—Assume 1100 people in the building on a zero day and $Q' =$

2000000. Then $H' = 1483251 + 1.27 \times 2000000 = 4023251$, with equal distribution of air, $t = 111$, and

$$R = \frac{4023251}{8.5\left(227 - \frac{111 + 0}{2}\right)} = 2758 \text{ sq. ft.} = 8274 \text{ lin. ft. coils.}$$

APPLICATION 3. *Where heating and ventilation are separate. Split system.*—With the same number of people in the building as given in Application 2, the heat loss H may be supplied by direct radiation and H_v by fan coils. In this case

$$R_v = \frac{2540000}{8.5\left(227 - \frac{80 + 0}{2}\right)} = 1597 \text{ sq. ft.} = 4791 \text{ lin. ft. coils.}$$

Final temperature 80° ($V = 1200$) gives three sections of regular vento (12 rows of coils) at 0° outside (See Table XXVI). This gives opportunity of reducing the direct radiation to give $t = 60°$ (See also Art. 140). In applying Equation 65, t is usually considered 140°. Conditions may exist, however, when this should change. For illustration, suppose that in a certain building most of the rooms are to be ventilated and that these rooms will have large amount of air delivered at low temperatures (Application 2). In such a case it may be economy to raise the air for *all* rooms to the lower temperature and supply more air to those rooms that would otherwise be heated with air at 140°, than to put in a heater large enough to heat all the air to 140° and then dilute with large amount of cold air to lower the temperature. Again, suppose that a school building contains, in addition to the regular class rooms, laboratories, etc., and auditorium and gymnasium, the two together requiring an amount of air sufficient to justify a separate fan system (a condition which frequently exists). It would be economy to separate the heating system for these rooms from the rest of the building because of the comparatively short time the rooms are in use. When not in use either fan unit may be shut down without interfering with the other unit, thus approaching a higher operating efficiency.

138. Approximate Rules for Plenum Heating Surfaces:— The following approximate rules are sometimes used in checking up heating surface in the coils. These should be used with caution.

Rule 1. "Allow one lineal foot of 1-inch pipe for each 65 to 125 cubic feet of room space, 65 for office buildings, schools, etc., and 125 for shops and laboratories." Since this building (Fig. 151) has approximately 500000 cubic feet of room space, it gives 7700 lineal feet of 1-inch pipe in the heater.

Rule 2.—"Allow 200 lineal feet of 1-inch pipe for each 1000 cubic feet of air per minute at a velocity of 1500 feet per minute." Applying to the above building when the air moves over the coils at 1000 feet per minute, the heated surface is only about four-fifths as valuable and would require 250 lineal feet per each 1000 cubic feet of air per minute. This gives 8333 lineal feet of coils.

139. Arrangement of Coils in Pipe Heaters:—Coil sections are arranged in two, three and four rows of pipes per section. Unless special reference is made to this point, the latter value is understood. Having found R for the heater, obtain from the temperature tables the number of pipe rows or sections deep the heater will need to be to produce the desired $t°$. Next, find the net wind area across the coils by dividing the total air moved by the assumed velocity of the coils. From the net wind area find the gross cross sectional area of the heater by the relation commonly used by manufacturers—

Gross wind area = 2.5 times net wind area.

From the gross area the size of the heater may be selected.

Application 1.—In Art. 137 let $R = 2222$, $Q = 1156935$, $V = 1000$ (deep heater) and $t = 140$. From Table XXV the heater will require 24 rows of coils in depth to give the required temperature. The net area will be $1156935 \div (60 \times 1000) = 19.3$ sq. ft., the gross area will be $2.5 \times 19.3 = 48.25$ sq. ft. and the heater size will probably be selected 6 ft. x 8 ft. Check for R by second method, following Application 3.

Application 2.—Let $R = 2758$, $Q' = 2000000$, $V = 1000$ (deep heater) and $t = 111$. Find the heater 16 + say 18 rows deep; net area = 33.3 sq. ft.; gross area = 83.3 sq. ft.; and size of heater = 9 ft. x 9 ft. or 2 divisions each 6 ft. x 7 ft. Check for R as in Application 1.

Application 3.—Let $R = 1597$, $Q' = 2000000$, $V = 1200$ (shallow heater) and $t = 80$. Find the heater 12 rows deep; net area = 27.8 sq. ft.; gross area = 69.5 sq. ft.; and size of heater 8 ft. x 8.75 ft.

A *second method* of obtaining pipe coil heater sizes is as follows: having found R, obtain the square feet of heating surface in any one row of coils across the heater by dividing R by the number of rows in depth (in the applications, 24, 18 and 12). Then from the usual relations existing between net area, gross area and heating surface per row obtain the size of the heater. Let the net area between the tubes, *N. A.*, the space occupied by the tubes, *T. A.*, and the gross cross sectional wind area through the tube, *G. W. A.*, be respectively

$$N.\ A. = \frac{Q \text{ or } Q'}{60\ V};\ T.\ A. = \frac{Q \text{ or } Q'}{40\ V};\ \text{and } G.\ W.\ A. = \frac{Q \text{ or } Q'}{24\ V} \qquad (69)$$

Since the cross sectional space *T. A.* occupied by the tubes is to the coil surface per row as 1 : 3.1416, the total coil surface in one row of tubes is

$$R_1 = \frac{3.1416\ (Q \text{ or } Q')}{40\ V} = .08\ \frac{(Q \text{ or } Q')}{V}.$$

Reduced to the basis of the net area, *N. A.*, we have

$$R_1 = 4.8 \text{ times } N.\ A. \qquad (70)$$

For illustration, when substitution is made in the three applications just cited we have: *first*, R (per row) = 92.6, *N. A.* = 92.6 ÷ 4.8 = 19.3, *G. A.* = 2.5 × 19.3 = 48.25 and the size of the heater is 6 ft. x 8 ft.; *second*, R (per row) = 153, *N. A.* = 32, *G. A.* = 80 and the size of the heater is 9 ft. x 9 ft.; *third*, R (per row) = 133, *N. A.* = 28, *G. A.* = 70 and the size of the heater is 8 ft. x 8.75 ft.

Slight variations occur in checking between the two methods because of the difficulty in selecting the exact number of rows of coils to fit the final temperatures. Either of the two methods as shown above for determining the size of the coil heater will give good practical results.

Assembled sections of pipe coil heaters are supplied by manufacturers from the smallest size of 3 ft. x 3 ft., to the largest size of 10 ft. x 10 ft., these dimensions being those of the gross cross sectional area, and not overall dimensions. Between the two limits, both height and breadth usually vary by 6 inch increments. For exact sizes, consult dimension tables in manufacturers' catalogs.

140. Arrangement of Sections and Stacks in Vento Cast Iron Heaters:—Referring to Applications 1, 2 and 3, Art. 139, find the number of stacks deep, the number of stacks high,

the number of sections to the stack and the size of the heater as compared with each application in coil heaters.

APPLICATION 1.—$R = 2222$ and $N. A. = 19.3$. From Table XXVI the number of stacks deep (regular, $t = 140$ and $t_o = 0$) = 6. From Table 53, Appendix, either of the following arrangements will give the required $N. A$: (a) one stack high, 60 in. section, 21 sections wide; size of heater 105 in. wide x 60 in. high x 55 in. deep. Check $R = 336 \times 6 = 2016$ sq. ft.; (b) a 40 in. stack above a 50 in. stack, 14 sections wide; size of heater 70 in. x 90 in. x 55 in. Check $R = (189 + 150.5) \times 6 = 2037$ sq. ft.; (c) a 40 in. stack above a 40 in. stack, 16 sections wide; size of heater 80 in. x 80 in. x 55 in. Check $R = 2 \times 172 \times 6 = 2064$ sq. ft.

APPLICATION 2.—$R = 2758$ and $N. A. = 33.3$. From the same tables (regular, $t = 111$ and $t_o = 0$) the number of stacks deep = 5 and the following arrangements will give the required $N. A.$: (a) a 60 in. stack above a 60 in. stack, 18 sections wide; size of heater 90 in. x 120 in. x 46 in. Check $R = 2 \times 288 \times 5 = 2880$ sq. ft.; (b) a 40 in. stack above a 60 in. stack, 21 sections wide; size of heater 105 in. x 100 in. x 46 in. Check $R = 2808$ sq. ft.; (c) a 40 in. stack above a 50 in. stack, 23 sections wide; size of heater 115 in. x 90 in. x 37 in. Check $R = 2789$ sq. ft.

APPLICATION 3.—$R = 1597$ and $N. A. = 27.8$. Regular sections, $t = 80$ and $t_o = 0$, find number of stacks deep = 3 and the following arrangements of heaters: (a) a 50 in. stack above a 60 in. stack, 17 sections wide; size of heater 85 in. x 110 in. x 28 in. Check $R = 1505$ sq. ft.; (b) a 40 in. stack above a 60 in. stack, 19 sections wide; size of heater 95 in. x 100 in. x 28 in. Check $R = 1525$ sq. ft.; (c) a 40 in. stack above a 50 in. stack, 21 sections wide; size of heater 105 in. x 90 in. x 28 in. Check $R = 1527$ sq. ft.

Other arrangements than the above may be made to suit the space set aside for the heater in the building plan. It will be noticed that the vento coils as taken from the tables check somewhat below the calculated R. Where space will permit add enough sections to the width of the heater to keep the total surface up to the calculated R and permit the velocity of the air over the coils to drop correspondingly.

141. Use of Hot Water in Indirect Coils:—In most cases low pressure steam is used as a heating medium in indirect heaters. It is possible to use hot water where a good supply

is available. In such an arrangement the coils will be calculated from Equation 65, using all values the same as for steam excepting t_s, which will be replaced by the average temperature of the water. The piping connections and the arrangement of the coils will follow the same general suggestions as already stated for direct heating.

142. Pounds of Steam Condensed per Square Foot of Heating Surface per Hour:—From Art. 137 the number of pounds of condensation per hour per square foot of surface in the coils is

$$m = \frac{H'}{R \times \text{heat given off per pound of condensation}} \tag{71}$$

APPLICATION.—Let $R = 2758$ and $H' = 4023251$; also let 1 pound of dry steam at 5 pounds gage in condensing to water at 212° give off $1156 - 180 = 976$ B. t. u. Then

$$m = \frac{4023251}{2758 \times 976} = 1.5 \text{ pounds.}$$

This amount is an average of all the coils. The first and last sections in any bank may vary above or below this amount 33 per cent. The first coils will condense as much as 2 pounds of steam per square foot of surface per hour under heavy service.

143. Pounds of Dry Steam Needed in Excess of the Exhaust Steam Given Off from the Engine:—In all steam driven plenum systems it is economy to use the exhaust steam from the power unit as a partial supply for the coils. Let the heating value of the exhaust steam from the engine be 85 per cent. of that of good dry steam (See Arts. 164 and 172); also let the engine use 40 pounds of dry steam per horsepower hour in driving the fan. From Art. 153 the engine will use $40 \times 13.7 = 548$ pounds of steam per hour and the heating value will be $976 \times .85 = 830$ B. t. u. per pound or $830 \times 548 = 454840$ B. t. u. total per hour. Then $4023251 - 454840 = 3568411$ B. t. u. and $3568411 \div 976 = 3657$ pounds of steam. The boiler will then supply to the engine and coils $3657 + 548 = 4205$ pounds of steam total and will represent $4205 \div 30 = 140$ boiler horse-power.

CHAPTER XII.

MECHANICAL WARM AIR HEATING AND VENTILATION. FAN-COIL SYSTEMS.

PRINCIPLES OF THE DESIGN, CONTINUED. FANS AND FAN DRIVES.

144. Theoretical Air Velocity:—The theoretical velocity of air flowing from any pressure p_a to any pressure p_b, is obtained from the general equation $v = \sqrt{2gh}$, where v is given in feet per second, $g = 32.16$ and h = head in feet producing flow. The latter value may be easily changed from feet of head to pounds pressure and vice versa.

When exhausting air from any enclosed space into another space containing air at a less density, the force which causes movement of the air is $p_a - p_b = p_x$. Pressures may be taken by any standard type of pressure gage. These show pressures above the atmosphere. When exhausting from any container into the atmosphere, $p_b = o$ and $p_a = p_x$. The fact that a difference of pressure exists between two points in air transmission indicates that there are two actual columns (or equivalent as in Fig. 10) of air at different densities connected and producing motion, or that by mechanical means a pressure difference is created which may be reduced to an equivalent head h by dividing the pressure head by the density of the air. Thus

$$h = \frac{\text{pressure difference}}{\text{density}} = \frac{p_a - p_b}{d}$$

Let $p_a - p_b = p_x$ = ounces of pressure per square inch of area producing velocity of the air; also, let g = acceleration due to gravity = 32.16 and d = density (weight of one cubic foot) of dry air at 60° and at atmospheric pressure (Table 7, Appendix). Substituting in the general equation

$$v = \sqrt{\frac{64.32 \times 144 p_x}{.0764 \times 16}} = 87\sqrt{p_x} \qquad (72)$$

Since the pressure producing flow is usually measured in inches of water, h_w, the above may be changed to read in equivalent height of water column by

$$h = \frac{\text{weight of water per cu. ft. at given temp.} \times h_w}{\text{weight of air at given temperature} \times 12} \tag{73}$$

Applying this to dry air at 60° and water at the same temperature (Tables 7 and 9, Appendix, also Art. 29),

$$h = \frac{62.37\, h_w}{12 \times .0764} = 68\, h_w$$

which when substituted in the general equation gives

$$v = \sqrt{64.32 \times 68\, h_w} = 66.2 \sqrt{h_w} \tag{74}$$

Equation 73 between the temperatures of 50° and 70° gives results varying between $v = 65.5 \sqrt{h_w}$ for 50° and $v = 66.5 \sqrt{h_w}$ for 70°. Taking the average value

$$v = 66 \sqrt{h_w} \tag{75}$$

Stated as a rule for approximate calculations *the theoretical velocity of air, when measured by a water column gage that measures in inches of water, equals sixty-six times the square root of the height of the column in inches.*

For calculations requiring greater accuracy, Equations 72 and 74 should take into account the density of the air and its drop in temperature. First, considering only density, let the pressure of one atmosphere at sea level be 29.92 inches of mercury (14.7 pounds = 235 ounces per square inch area). Since the density is proportional to the absolute pressure, the temperature remaining constant, we have from Equation 72 with air exhausting into the atmosphere

$$v = \sqrt{\frac{64.32 \times 144\, p_x}{.0764 \times 16 \times \dfrac{235 + p_x}{235}}} = 1336 \sqrt{\frac{p_x}{235 + p_x}} \tag{76}$$

Also from the relation existing between Equations 72 and 74, Equation 76 reduces to

$$v = 1336 \sqrt{\frac{h_w}{407 + h_w}} \tag{77}$$

Second, considering both density and temperature, Equations 76 and 77 become

$$v = 1336 \sqrt{\left(\frac{460 + t}{520}\right)\frac{p_x}{235 + p_x}} \tag{78}$$

$$v = 1336 \sqrt{\left(\frac{460 + t}{520}\right)\frac{h_w}{407 + h_w}} \tag{79}$$

To facilitate calculation, the second columns in Tables XXX and XXXI have been compiled from Equations 76 and 77 respectively, and the second column in Table XXXII has been compiled for different temperatures on the basis of 60° from that portion of Equations 78 and 79 included within the parenthesis. From these three columns of tabulations the theoretical velocity of air under any pressure and temperature change may be obtained without using the equations, by multiplying the velocities found in Tables XXX and XXXI by the factor for temperature correction given in Table XXXII. Other points of information concerning velocities,

TABLE XXX.

Column 2 figured from Equation 76.

Pressure in ounces per sq. in.	Velocity of dry air at 60° escaping into the atmosphere through any shaped orifice in any pipe or reservoir in which a given pressure is maintained. Ft. per sec.	Ft. per min.	Vol. of air in cu. ft. which may be discharged in 1 min. through an orifice having an effective area of discharge of 1 sq. in. Col. 3 ÷ 144	H. P. required to move the given vol. of air under the given conditions of discharge. (Col. 3 × Col. 1) / (16 × 33000)
1/8	30.80	1848.00	12.83	0.00044
1/4	43.56	2613.60	18.15	0.00124
3/8	53.27	3196.20	22.19	0.00227
1/2	61.56	3693.60	25.65	0.00349
5/8	68.79	4127.40	28.66	0.00489
3/4	75.35	4521.00	31.47	0.00642
7/8	81.37	4882.20	33.90	0.00809
1	86.97	5218.20	36.24	0.00988
1 1/8	92.18	5530.80	38.41	0.01178
1 1/4	97.18	5830.80	40.49	0.01380
1 3/8	101.90	6114.00	42.46	0.01592
1 1/2	106.40	6384.00	44.33	0.01814
1 5/8	110.82	6649.20	46.11	0.02046
1 3/4	114.86	6891.60	47.86	0.02284
1 7/8	118.85	7131.00	49.52	0.02533
2	122.47	7348.20	51.03	0.02787

pressures, weights and horse-powers in moving air may be obtained by multiplying by the factors as given in the respective columns.

TABLE XXXI.

Column 2 figured from Equation 77.

Pressure head in inches of water	Velocity of dry air at 60° escaping into the atmosphere through any shaped orifice in any pipe or reservoir in which a given pressure is maintained.	
	Feet per second	Feet per minute
.1	20.94	1256.40
.2	29.67	1780.20
.3	36.25	2175.60
.4	41.86	2511.60
.5	46.80	2708.00
.6	51.26	3075.60
.7	55.36	3321.60
.8	59.10	3546.00
.9	62.60	3756.00
1.	66.14	3968.40
1.1	69.36	4161.60
1.2	72.44	4346.40
1.3	75.39	4523.40
1.4	78.21	4692.60
1.5	80.96	4857.60
1.6	83.59	5015.40
1.7	86.16	5169.60
1.8	88.65	5319.00
1.9	91.27	5476.20
2.	93.42	5605.20
2.1	95.72	5743.20
2.2	97.96	5877.60
2.3	100.15	6009.00
2.4	102.29	6137.40
2.5	104.39	6263.40
2.6	106.43	6385.80
2.7	108.46	6507.60
2.8	110.43	6625.80
2.9	112.37	6742.20
3.	114.28	6856.80
3.1	116.15	6969.00
3.2	118.00	7080.00
3.3	119.81	7188.60
3.4	121.60	7296.00
3.5	123.36	7401.60

APPLICATION 1.—Air is exhausting from an orifice in an air duct into the atmosphere. The pressure of the air within the duct is one ounce by pressure gage or 1.74 inches of

water by Pitot tube. Assuming the air to be dry and the barometer 29.92 inches when the water in the tube and the air current are 60°, what is the theoretical velocity of the air?

Solution.—By Tables XXX and XXXI, $v = 86.97$. Check this by Equation 76.

Application 2.—In Application 1 let the duct pressure and temperature be 3 inches of water and 70° respectively. What is the theoretical orifice velocity?

Solution.—From Table XXXI, $v = 114.28$ at 60°. Multiplying this by 1.01 from Table XXXII = 115.42 F. P. S. velocity. Check by Equation 79.

TABLE XXXII.

Temp. in degrees.	Factor for relative vel. at same pressure also relative powers to move same vol. of air at same vel. = $\sqrt{\dfrac{\text{Wt. at any T}}{\text{Wt. at } 460° + 60°}}$	Factor for relative pressure, also wt. of air moved at same vel. = $\dfrac{460° + 60°}{T}$	Factor for relative vel. to move same wt. of air also relative pressure to produce the vel. to move same wt. of air = 1 ÷ Col. 3	Factor for relative power to move same wt. of air at vel. in column 4 and pressure in column 4 = factor in column 4 squared
30	.97	1.07	.93	.87
40	.98	1.04	.96	.92
50	.99	1.02	.98	.96
60	1.00	1.00	1.00	1.00
70	1.01	.98	1.02	1.04
80	1.02	.96	1.04	1.08
90	1.03	.94	1.06	1.13
100	1.04	.92	1.09	1.19
125	1.06	.89	1.12	1.25
150	1.08	.85	1.18	1.39
175	1.10	.82	1.22	1.49
200	1.13	.79	1.27	1.61
250	1.17	.73	1.37	1.88
300	1.21	.68	1.47	2.16
350	1.25	.64	1.56	2.43
400	1.28	.60	1.67	2.79
500	1.36	.54	1.85	3.42
600	1.43	.49	2.04	4.16
700	1.49	.45	2.22	4.93
800	1.56	.41	2.44	5.95

145. Actual Amount of Air Exhausted:—When air at any pressure is exhausted from one receptacle to another through an orifice, nozzle or short pipe, the actual velocity is reduced below the theoretical velocity and the effective

area of the jet or stream is less than the actual area of the opening. These variations are due to the shape of the inlet and the friction on the contact surface. To find the amount of air Q moved per second, multiply the theoretical velocity by the actual area and by a constant which is the product of the coefficient of reduced velocity and the coefficient of reduced area. $Q = C\,v\,A$, where C = constant, usually called *coefficient of efflux.* Kent, page 615, quotes from experiments by Weisbach the following values:

For conoidal mouthpiece, of form of the contracted vein, with pressures of from 0.23 to 1.1 atmospheres	C = 0.97 to 0.99
Circular orifice in thin plate	C = 0.56 to 0.79
Short cylindrical mouthpiece	C = 0.81 to 0.84
Short cyl. mouthpiece rounded at the inner end	C = 0.92 to 0.93
Conical converging mouthpiece	C = 0.90 to 0.99

146. Results of Tests to Determine the Relation between Pressure and Velocity in Air Transmission:—In fan construction the number and shape of the blades, the sizes of the inlet and outlet openings, the shape and size of the casement around the wheel and the speed, all have an effect upon the relation between the pressure and the velocity of the air discharge. From tests conducted by the author, the curves shown in Fig. 148 were obtained. A No. 2 Sirocco

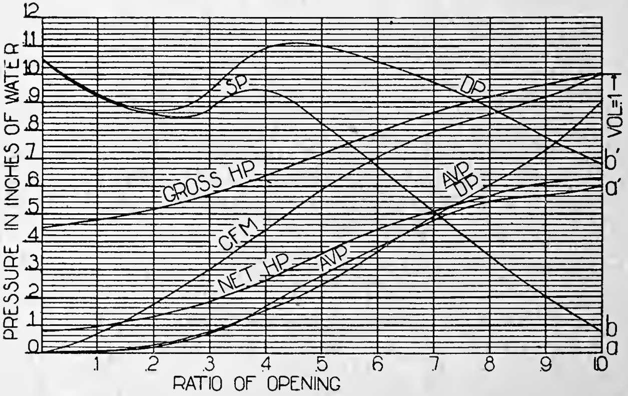


Fig. 148.

blower was belted to an electric motor and delivered air to a horizontal, circular pipe whose length was nine times the diameter. This pipe was provided with reducing nozzles which varied the area of discharge by tenths from full opening to full closed. The air tube was provided also with manometer tubes for static, dynamic and velocity pressures, also, an adjustable scale reading in two positions, either .01 or .002 inch of water. The gross power was taken by wattmeter and the delivered power from motor to fan was taken by dynamometer. In addition to this, the frictional horsepower of the fan and motor unit was obtained by removing the fan wheel from the shaft and taking readings with all other conditions remaining as nearly constant as possible. The frictional power, when deducted from the gross power recorded by the wattmeter, gave the readings for the net horse-power curve. A galvanized iron intake, enlarged from the size of the fan intake to a rectangular four square feet in area and divided by fine wires into squares to the size of the standard anemometer, was used to find the volume of air moved per minute. This volume is shown in the curve C. F. M. To check the curve, the volume was calculated for each opening by the Pitot tubes on the side of the experimental pipe.

To fully understand this article, refer to Art. 29 and note that *A*, Fig. 12, registers *static pressure plus velocity pressure.* This sum may be called the *dynamic pressure.* Also note that *B* registers *only static pressure*, i. e., that pressure which acts equally in all directions and serves no usefulness in moving the air. Also, note that $A - B = C$, i. e., dynamic pressure minus static pressure equals velocity pressure. When applied in the form shown by *C*, the pressure recorded is that due to the velocity only. This is the form commonly used. Referring again to Fig. 148, A V P is that pressure recorded by *C* when applied to the air current at the fan outlet, = air velocity pressure. P V P is that pressure (obtained by Equations 72 and 77) that would be shown on *C* if the air were moving as fast as the tip of the blades on the fan wheel, = peripheral velocity pressure. P V P = 1 in Fig. 148. D P is the dynamic pressure and would be found by applying *A* only. S P is the static pressure as stated above.

In the tests, the fan was run at constant speed and the dynamic, static and velocity pressures were measured about

midway of the pipe at full opening. Then the openings were changed by 10 per cent. reductions until the pipe was fully closed and similar readings taken for each reduction. These readings were plotted in the upper set of curves. Because the manometer tubes were located some distance from the end of the experimental pipe, there was a static pressure, *ab*, recorded at full opening. This caused the dynamic pressure to be raised a corresponding amount, *a′ b′*. If the tubes had been located at the delivery end of the pipe the static and dynamic pressures would have fallen from *b* and *b′* to *a* and *a′*. The peripheral velocity of the wheel was 2828 feet per minute and the corresponding pressure, with corrections for temperature, was found by Equation 75 to be .5 inch of

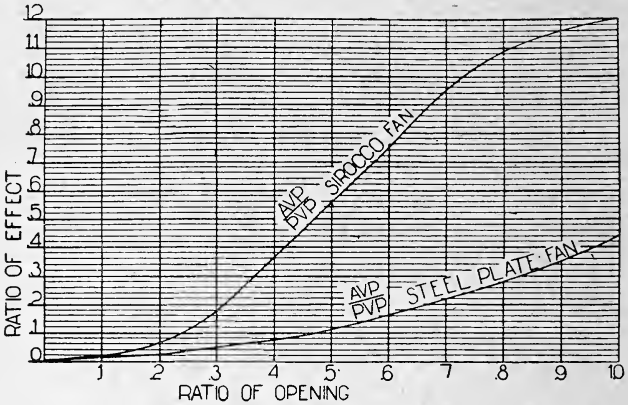


Fig. 149.

water. The relation between the peripheral velocity pressure and the air velocity pressure is shown in the upper curve, Fig. 149. In applying this to fan practice it shows the relation between the velocity of a point on the wheel circumference and that of the air leaving the wheel. Notice that at full opening and discharging into free air, A V P : P V P :: 1.2 : 1. Since the velocities vary as the square roots of the pressures ($v = \sqrt{2gh}$), we find the velocities to be $\sqrt{1.20} : \sqrt{1} = 1.1 : 1$. That is to say, for this fan the air velocity at the free opening of the fan is 1.1 times

the peripheral velocity of the wheel. The corresponding velocity of air from a steel plate fan as reported by the American Blower Co. and as shown on the lower chart, is $\sqrt{.45} : \sqrt{1} = .67 : 1$, or .61 of the speed of the Sirocco fan for the same wheel speed. The resistance offered by the ducts in the average plenum heating system is equivalent, we will say, to that offered by a 75 per cent. gate opening in the experimental pipe. According to the diagrams for this opening, the ratio A V P to P V P is 1.04 for the Sirocco fan and .25 for the steel plate fan. The ratio of the air velocities to the peripheral velocities then are, respectively, $\sqrt{1.04} : \sqrt{1} = 1.02 : 1$ and $\sqrt{.25} : \sqrt{1} = .5 : 1$. These show that with a 75 per cent. opening and with the fan wheels running with a peripheral velocity of 3000 feet per minute, the air would be entering the ducts at $1.02 \times 3000 = 3060$, and $.5 \times 3000 = 1500$ feet per minute respectively for the two types. Conversely, if it were desired to have the air enter the ducts at 1500 feet per minute, with a resistance equivalent to a 75 per cent. opening, the fan wheels would have peripheral speeds of $1500 \div 1.02 = 1470$, and $1500 \div .5 = 3000$ feet per minute respectively. From these velocities may be obtained the wheel diameter for any given R. P. M. Other models of the Sirocco and multiple blade type of fans will show different characteristics than the one under consideration. It will be seen from the above analysis that the change in construction from the steel plate type to the multiblade type permits a smaller wheel and fan to be installed for any given work desirable. From Equation 86 it is seen that the power required to drive a fan varies as the fifth power of the diameter and as the cube of the speed. With a given amount of air, Q, required per minute, the power will be diminished by reducing the diameter of the wheel or by reducing the speed of the fan. Manufacturers' catalogs should be consulted for capacities, sizes, etc. Such tables are supplied by the trade in form for easy reference and use.

147. Work Performed and Horse-Power Consumed in Moving Air:—The foot pounds of work performed in moving air equals the product of the moving force into the distance moved through in any given time. Let $p_a - p_b = p_x =$ moving force of air in ounces per square inch and $A =$ cross-sectional area of air stream in square inches. Then the

pounds per square inch will be $p_x \div 16$, and the foot pounds of work, W, and the horse-power, H. P., absorbed per minute by the air will be

$$W = \frac{60\, p_x\, A\, v}{16} = 3.75\, p_x\, A\, v \qquad (80)$$

$$H.\ P. = \frac{3.75\, p_x\, A\, v}{33000} = .000114\, p_x\, A\, v \qquad (81)$$

Equation 81 may be stated in terms of the cubic feet or air discharged per minute. Take the relation between p_x and h_w at 60° as $12\, p_x = 16 \times .433\, h_w$; also the relation $A\, v = 144\, Q$ when Q = cubic feet of air discharged per second and, from Equation 75, $h_w = v^2 \div 4356$. Then by substituting in Equation 81

$$H.\ P. = \frac{3.75 \times .577 \times v^2 \times 144\, Q}{4356 \times 33000} = .0000022\, v^2\, Q \qquad (82)$$

In Equations 80 to 82, p_x = total pressure drop in system being investigated (dynamic pressure), and v = velocity corresponding to p_x.

ILLUSTRATION.—In a plenum heating and ventilating system the pressure above atmosphere at the fan outlet (gage pressure, corresponding to resistance of ducts and heater coils) is .6 inch of water; the pressure below atmosphere at the fan inlet (resistance of tempering coils and air inlet) is .15 inch of water; the equivalent velocity head is .25 inch of water; then the pressure the fan is working against is 1 inch of water = .58 ounce = p_x.

APPLICATION 1.—The constant area of a stream of dry air at 60° exhausting between the pressures of $p_a = 1\frac{1}{2}$ ounces and $p_b = \frac{1}{2}$ ounce, is 400 square inches. What is the work performed per minute and the horse-power consumed? (For velocity see second column Table XXX),

$W = 3.75 \times (1\frac{1}{2} - \frac{1}{2}) \times 400 \times 86.97 = 130500$ foot pounds, and $H.\ P. = .000114 \times (1\frac{1}{2} - \frac{1}{2}) \times 400 \times 86.97 = 3.96$.

APPLICATION 2.—A fan is delivering 1000000 cubic feet of air per hour to a heating system at a temperature of 100° and with a total pressure of ¾ ounce. What is the theoretical horse-power of the fan? From Tables XXX and XXXII, $v = 75.35 \times 1.04 = 78.36$ and

$$H.\ P. = .0000022 \times (78.36)^2 \times 278 = 3.76$$

The actual horse-power of a blower fan is the horse-power absorbed in moving the air plus the horse-power absorbed by the blower. Let E = efficiency of the blower. Then Equations 81 and 82 become

$$H.\ P. = \frac{.000114\ p_x\ A\ v}{E} \tag{83}$$

$$H.\ P. = \frac{.0000022\ v^2\ Q}{E} \tag{84}$$

The value of E changes with the type of fan. In the steel plate fan it will vary from 20 to 40 per cent. Average 30 per cent. In the Sirocco and multiblade fans it will vary from 50 per cent. at 50 per cent. rated capacity to 70 per cent. at 100 per cent. rated capacity. The latter value is safe (See also Art. 152).

148. Carpenter's Practical Rules for Fan Capacities:— Professor Carpenter in H. & V. B. has summarized tests on steel plate fans as follows:

Rule.—"The capacity of fans, expressed in cubic feet of air delivered per minute, is equal to the cube of the diameter of the fan wheel in feet multiplied by the number of revolutions multiplied by a coefficient having the following approximate value: for fan with single inlet delivering air without pressure, 0.6; *delivering air with pressure of one inch,* 0.5; *delivering air with pressure of one ounce,* 0.4; *for fans with double inlets, the coefficient should be increased about* 50 *per cent. For practical purposes of ventilation, the capacity of a fan in cubic feet per revolution will equal* .4 *the cube of the diameter in feet."*

Rule.—"The delivered horse-power required for a given fan or blower is equal to the 5th power of the diameter in feet, multiplied by the cube of the number of revolutions per second, divided by one million and multiplied by one of the following coefficients: for free delivery, 30; *for delivery against one ounce pressure,* 20; *for delivery against two ounces of pressure,* 10."

Stated as equations these rules are as follows:

$$D = \sqrt[3]{\frac{\text{Cu. ft. of air per min.}}{C \times R\,P\,M}} \tag{85}$$

where D = the diameter in feet and C = the coefficient, .4

for pressure of one ounce, .5 for pressure of one inch, and .6 for no pressure.

$$H.\,P. = \frac{D^5\,(R\,P\,S)^3 \times C}{1000000} \tag{86}$$

where $C = 30$ for open flow, 20 for one ounce and 10 for two ounces pressure respectively.

Note.—In using Equations 85 and 86 for Sirocco or multivane fans, C should be 1.1, 1.2 and 1.3 for 85, and 100, 95 and 90 for 86.

149. Approximate Fan Sizes:—Table XXXIII gives sizes of important features in fan casing, wheel and openings referred to the wheel diameter.

TABLE XXXIII.

	Steel plate fan	Multiblade fan
Diameter of wheel	D	D
Diameter of inlet, single......	.60 D to .70 D	1.0 D to 1.2 D
Dimensions of exhaust	.50 D to .60 D	.6 D to .8 D by .7 D to 1.0 D
Wheel width inlet circum...	.50 D to .60 D	
Wheel width outer circum...	.35 D to .45 D	.5 D

Type of fan	Space occupied Full housed	Discharge vert.	Discharge horiz.
Steel plate............		1.7 D to 1.5 D	1.5 D to 1.7 D
Multiblade	Length	1.8 D to 2.0 D	1.4 D to 1.6 D
Steel plate............		1.5 D to 1.7 D	1.7 D to 1.5 D
Multiblade	Height	1.4 D to 1.6 D	1.8 D to 2.0 D
Steel plate............		.7 D to 1.2 D	.7 D to 1.2 D
Multiblade	Width	1.3 D to 1.5 D	1.3 D to 1.5 D

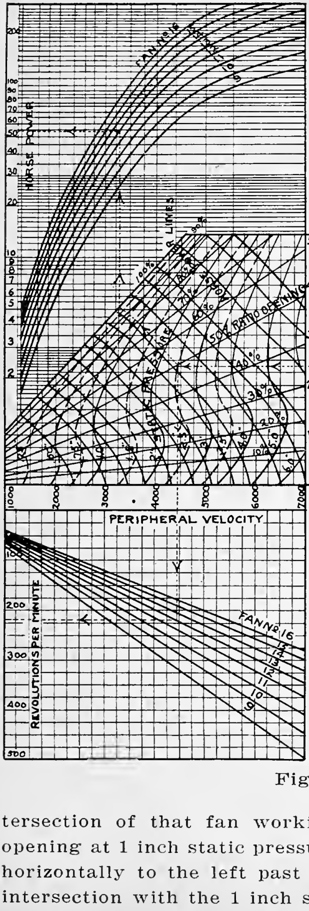
150. Selection of Fan for Given Capacity:—*By calculation.*—(See Art. 153, Application 1).

By graphical analysis.—Assume the conditions given in Art. 153, Application 1, and apply Fig. 150 as follows: locate 33330 on the C F M scale, rise vertically from this point to the in-

EXAMPLE SHOWING APPLICATION OF CURVES.

PROBLEM. Determine size of fan, revolutions per minute and brake horse-power required to deliver 80,000 cubic feet per minute against a static pressure of 2.5″ W. G.

SOLUTION. — Locate 80,000 on the C. F. M. scale and project vertically upward from this point to the intersection of that fan working nearest 50% ratio opening at 2.5″ S. P., which in this case is Fan No. 13 (follow dash lines on curves). From this point project horizontally to intersect 2.5″ S. P. curve and from thence upward parallel to horse-power lines to intersect Fan No. 13 at which point read 53.5 B. H. P. on horse-power scale to left. From same point on S. P. curve project vertically downward to intersect Fan No. 13 reading 218 R. P. M. on scale to left.



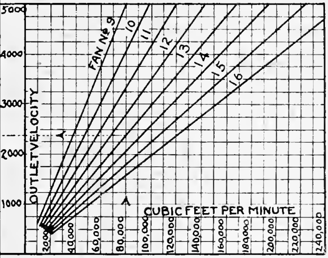


Fig. 150.

tersection of that fan working nearest 50 per cent. ratio opening at 1 inch static pressure and find a No. 10 fan. Move horizontally to the left past the outlet velocity 1700 to the intersection with the 1 inch static pressure curve. Call this

point *A*. From here drop vertically past the peripheral velocity 2850 feet per minute to the No. 10 slope and then move horizontally to the left to 180 revolutions per minute. Also, from *A* parallel the horse-power curve to the end, then rise to curve No. 10 and move horizontally to the left to 9 horse-power. Check the peripheral and outlet velocities, also other values found, by Table 57, Appendix.

Similar charts of workable size may be had from the manufacturers covering fans, Nos. 9 to 16 inclusive.

151. Fan Drives:—Fans for heating and ventilating purposes may be driven by simple horizontal or vertical, throttling or automatic steam engines, or by electric motors. In either engine or motor drives the power may be direct-connected or belt-connected to the fan. Direct-connected fan units make very neat arrangements but they require slow speed engines or motors and are frequently so large as to be prohibitive. Engine fans having poor attention are liable to pound, the noise carrying through the fan to the air current and up to the rooms. In belted drives engines or motors are run at higher speeds and are either set off from the fan 10 feet or more to get good belt contact or used with a tightener. Chain drives are sometimes installed. They are positive in speed, fairly quiet in operation, permit the same speed reductions as belt drives and economize floor space.

In deciding between engine or motor drives with steam coils, the steam from the engine should be considered a credit to the heating system since it is exhausted into the heater coils and used instead of live steam from the boilers. Engines of high efficiency are not essential when the exhaust steam can be used for heating. Enclosed engines running in oil are preferred for high speeds. Belt drives should have the tight side below to increase the arc of contact.

Electric motors should be specified for installations where exhaust steam can not be used, as in systems for ventilating only. They are more satisfactory in many ways than steam engines but are more expensive to operate. Direct current motors are desired in many places because of the convenience in obtaining speed changes and direct-connections. Alternating current motors operate at higher speeds, but may have speed reductions of 40 per cent. where required. When motors are specified the alternating current constant speed machine with belt drive is generally selected.

152. Speed of the Fan:—A blower fan, exhausting into the open air, will deliver air with a lineal velocity approximately that of the peripheral velocity of the fan blades. If this same fan is connected to a system of ducts and heater coils, the lineal velocity of the air is reduced because of the increased resistance in the duct system. This causes the air to *lag* or *slip* between the fan blades and the casing. In the average heating system using multiblade fans this slip may be as great as 30 per cent. (See Art. 146). It is sometimes convenient in applying blowers to heating systems to consider the lineal velocity of the air as it leaves the fan to be two-thirds that of the periphery of the fan blades. Since the velocity of the air upon delivery from the fan should not exceed 1800 to 2500 feet per minute, the outer point on the fan blades should not be expected to move faster than 2700 to 3750 feet per minute. Knowing this peripheral velocity, the revolutions per minute may be selected and the diameter obtained.

In direct-connected fans the revolutions per minute must agree with those of the attached engine or motor. In belted fans this restriction does not apply. Ordinary blower fans running at high speeds are noisy and practice has determined the number of revolutions to use. Table XXXIV gives speeds that may be recommended for such use.

TABLE XXXIV.

Speeds of Sirocco and Multiblade Blower Fans, in R P M

Diameter of wheel in inches	Differential pressures				
	.288 oz. .5 in.	.433 oz. .75 in.	.577 oz. 1 in.	.865 oz. 1.5 in.	1.154 oz. 2 in.
24	322	591	454	554	642
36	214	260	302	369	427
48	161	196	225	277	321
60	129	157	181	223	257
72	107	130	151	185	214
84	92	112	130	159	184
90	86	104	121	148	171
96	81	99	113	139	160

In recent developments in blower fans the number of blades has been increased and the depth of the blades has been diminished, making the operation of the fan somewhat similar to that of the steam turbine. These changes have

produced higher efficiencies under test than were possible with the ordinary steel plate or paddle wheel fan. As a result, fan sizes for given capacities have been reduced. Tables 55, 56 and 57, Appendix, give summaries of the latest catalog data.

153. Selection of the Engine and Fan:—Determine the power of the fan from Equations 83 or 84. Assume a certain ratio between this and the engine power, say as 3 is to 4, then

$$H.\ P.\ \text{of the engine} = \frac{4}{3}\ H.\ P.\ \text{of the fan} \qquad (87)$$

Having obtained the horse-power of the engine, find the size of the cylinder as follows: let p_a = absolute initial pressure of the steam in the cylinder, and r = number of the steam expansions in the cylinder = reciprocal of the per cent. of cut-off = the sum of the displacement at release plus the clearance divided by the sum of the displacement at cut-off plus the clearance. The cut-off allowed for high speed engines in power service approximates 25 per cent. stroke, but in engines for blower work this may be taken 50 per cent. stroke. Find the mean effective pressure, p_1, by the equation

$$p_1 = p_a \frac{1 + \text{hyperbolic logarithm of } r}{r} - \text{back pressure} \qquad (88)$$

Let l = length of the stroke in inches and N = number of revolutions per minute and apply the equation

$$H.\ P. = \frac{2\ p_1\ l\ A\ N}{12 \times 33000} \qquad (89)$$

and find A, the area of the cylinder, from which obtain d, the diameter of the cylinder. In applying Equation 89 it will be necessary to assume l. For engines operating blowers this may be taken

$$2\ l\ N = 200 \text{ to } 400$$

Equation 88 assumes that the steam in the cylinder expands according to the hyperbolic curve, $p\ v = p'\ v'$. For values of hyperbolic (Naperian) logarithms see Table 5, Appendix. It also assumes no loss in the recompression of the steam in the cylinder. Both assumptions are only approximately correct, but the errors are slight and to a certain degree, tend to neutralize each other, hence the final

results from this equation are near enough to be used for fan engine calculations. For such work as this, r may be taken from 2 to 3. The back pressure should not be higher than 5 pounds gage (19.7 pounds absolute). This is determined by the pressure in the coils carrying exhaust steam, which frequently drops to atmosphere or below.

In determining the cylinder diameter and length of stroke it may be necessary to make two or more trial applications before good sizes are obtained. When initial steam pressures are low, say not to exceed 30 pounds gage, mean effective pressures are small, thus requiring cylinders of large diameter. In such cases the diameter of the cylinder may be greater than the length of stroke. Where high pressure steam is used, say 100 pounds gage, the diameter of the cylinder will be less than the length of the stroke.

Application 1.—Assume the following to fit the design shown in Figs. 151, 152 and 153; dry steam to the engine at 100 pounds gage pressure; direct-connected unit; Sirocco type fan, single inlet, 60 per cent. efficiency, running at 180 revolutions per minute and delivering 2000000 cubic feet of air per hour to the building against a static pressure of 1 inch of water (total pressure 1.15 inch). (See Table 57, Appendix); steam cut-off in the cylinder at one-third stroke and used in the coils at 5 pounds gage pressure. Find the sizes and horse-powers of the engine fan unit.

From Equation 84, with v = velocity due to a total pressure of 1.15 inch of water,

$$\text{Fan } H.\ P. = \frac{.0000022 \times (71)^2 \times 555.5}{.60} = 10.26$$

From extended tables of the A. B. Co. similar to Table 57, Appendix, find a No. 10 fan, 60 inch wheel, 33650 C. F. M., 181 R. P. M., 10.2 H. P. Peripheral velocity of wheel 2845 F. P. M. Checking these values with Equations 85 and 86

$$D \text{ of fan} = \sqrt[3]{\frac{2000000}{60 \times 1.3 \times 180}} = 5.2 \text{ ft.} = 62 \text{ in.}$$

$$H.\ P. \text{ of fan} = \frac{(5.2)^5 \times (3)^3 \times 97}{1000000} = 10.1$$

From Equations 87, 88 and 89.

$$H.\ P. \text{ of engine} = \frac{4}{3} \times 10.26 = 13.68$$

$$p_1 = 115 \left(\frac{1 + 1.0986}{3}\right) - 19.9 = 60.5 \text{ lbs. per sq. in.}$$

If $2\ l\ N = 250$, $l = 250 \div 360 = .69$ ft. $= 8.25$ in. and

$$A = \frac{13.68 \times 12 \times 33000}{2 \times 60.5 \times 8.25 \times 180} = 30 \text{ sq. in.} = 6.25 \text{ in. diameter.}$$

The engine is 6.25 in. $\times$ 8.25 in., at 180 R. P. M.

Application 2.—Assume the values as in Application 1, excepting that the steam is taken from a conduit main at a pressure of 30 pounds per square inch gage, that $2\ l\ N = 300$, and that the steam cut off in the cylinder is at one-half stroke. As before, D of fan $= 5.2$ ft.; H. P. of fan $= 10.26$; and H. P. of engine $= 13.68$. The mean effective pressure is

$$p_1 = 45 \left(\frac{1 + .6931}{2}\right) - 19.9 = 18.2 \text{ lbs. per sq. in.}$$

$$A = \frac{13.68 \times 12 \times 33000}{2 \times 18.2 \times 10 \times 180} = 83 \text{ sq. in., and the size of}$$

the engine is 10.25 in. $\times$ 10 in., at 180 R. P. M.

154. Piping Connections Around Heater and Engine:—Where fans are run by steam power the steam supply pressure is higher than that in the coils and the live steam must enter the coils through a pressure reducing valve. Where this reduction is made to 5 pounds or below, the live steam may enter the same main with exhaust steam from the engine, the back pressure valve on the exhaust steam line providing an outlet to the atmosphere in case the pressure runs above the 5 pounds allowable back pressure. If the back pressure increases above 5 pounds, the efficiency of the engine is reduced. Where the condensation from the live steam is desired free from oil, a certain number of coils are tapped for exhaust steam and this condensation trapped to a waste or sewer, the other coils delivering to a receiver for boiler feed or other purposes as may be required.

Every heating system should be fully equipped with pressure reducing valves, back pressure valves, traps, and a sufficient number of globe or gate valves on the steam supply and gate valves on the returns to make the system flexible and responsive to varying demands. Supply and return connections for heater stacks should be the same as for the amount of direct radiation that will condense the same

amount of steam. Some engineers advocate a water-seal of 20 to 30 inches on the return end of each section, thus making each section independent in its action. Where the coils are very deep this is a benefit.

155. Application to School Buildings:—Figs. 151, 152 and 153, and Table XXXV show an application of plenum heating and ventilating to a school building. The table gives some of the calculated results. Most of the applications throughout Chapters X, XI and XII, also refer to this same building.

The plans show the double duct system with plenum chamber and ducts laid just below the basement floor. The small arrows show the heat registers and vent registers for each room. The same stack which serves as a heat carrier to the room on one floor serves as the vent stack for the corresponding room on the floor above, there being a horizontal cut-off between them. The cut-off at the heat register is curved to throw the current of heated air into the the room with the least possible friction or eddy currents.

TABLE XXXV.

Data Sheet for Figs. 151, 152, 153.

Room	Heat loss in B. t. u. per hour from room, not counting ventilation	Room	Heat loss in B. t. u. per hour from room, not counting ventilation	Room	Heat loss in B. t. u. per hour from room, not counting ventilation
1	51,520	11	81,130	21	81,130
2	74,200	12	126,973	22	17,150
3	29,400	13	44,583	23	113,800
4	36,260	14	60,907	24	17,150
5	42,210	15	70,224	25	35,189
6	35,350	16	50,862	26	53,438
7	-------	17	51,940	27	102,333
8	16,520	18	24,843	28	28,420
9	16,520	19	23,660	29	37,380
10	42,210	20	63,840	30	54,110
Totals	344,190	Totals	540,100	Totals	598,961

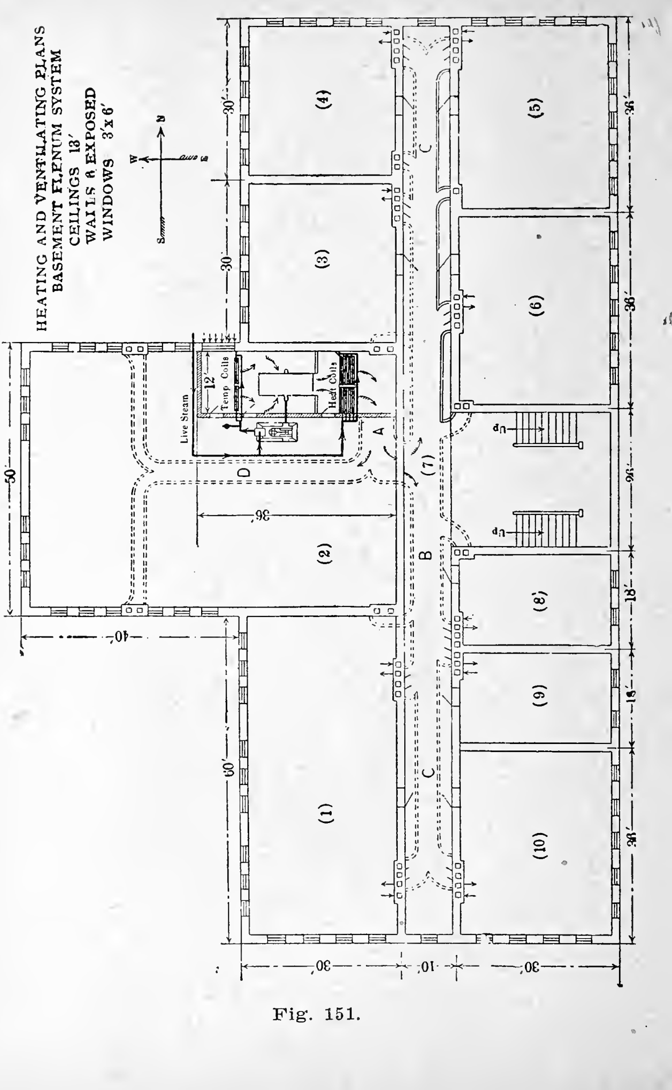

HEATING AND VENTILATING PLANS
BASEMENT PLENUM SYSTEM
CEILINGS 13'
WALLS & EXPOSED
WINDOWS 3'x 6'
Live Steam
Temp Coils
Heat Coils
Up
Up

Fig. 151.

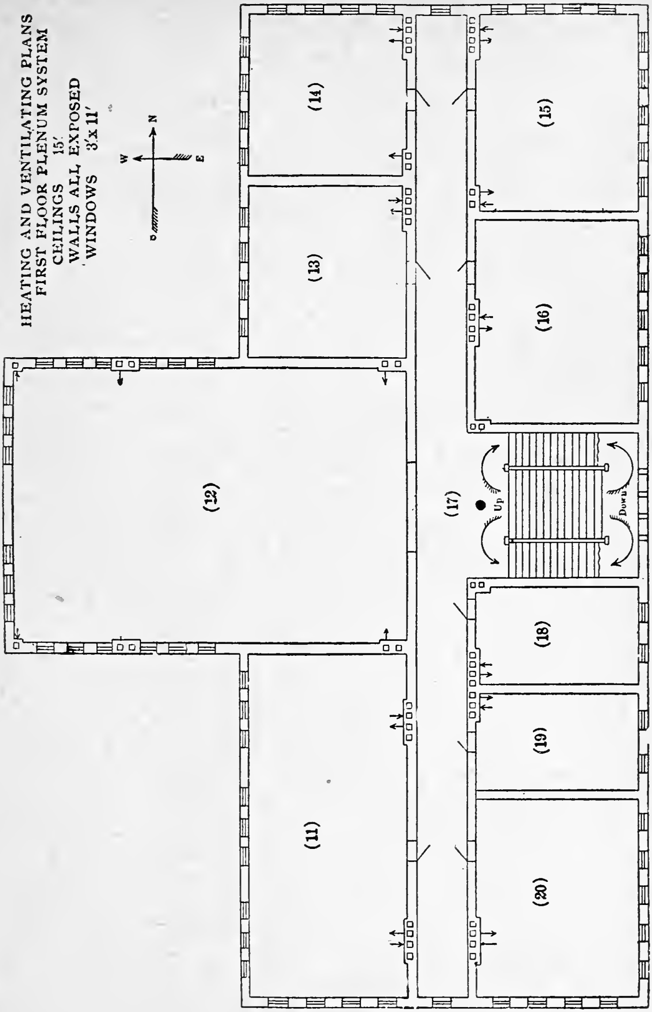

HEATING AND VENTILATING PLANS
FIRST FLOOR PLENUM SYSTEM
CEILINGS 15'
WALLS ALL EXPOSED
WINDOWS 3'x 11'
W
N
E
(11)
(12)
(13)
(14)
(15)
(16)
(17)
(18)
(19)
(20)
Up
Down

Fig. 152.

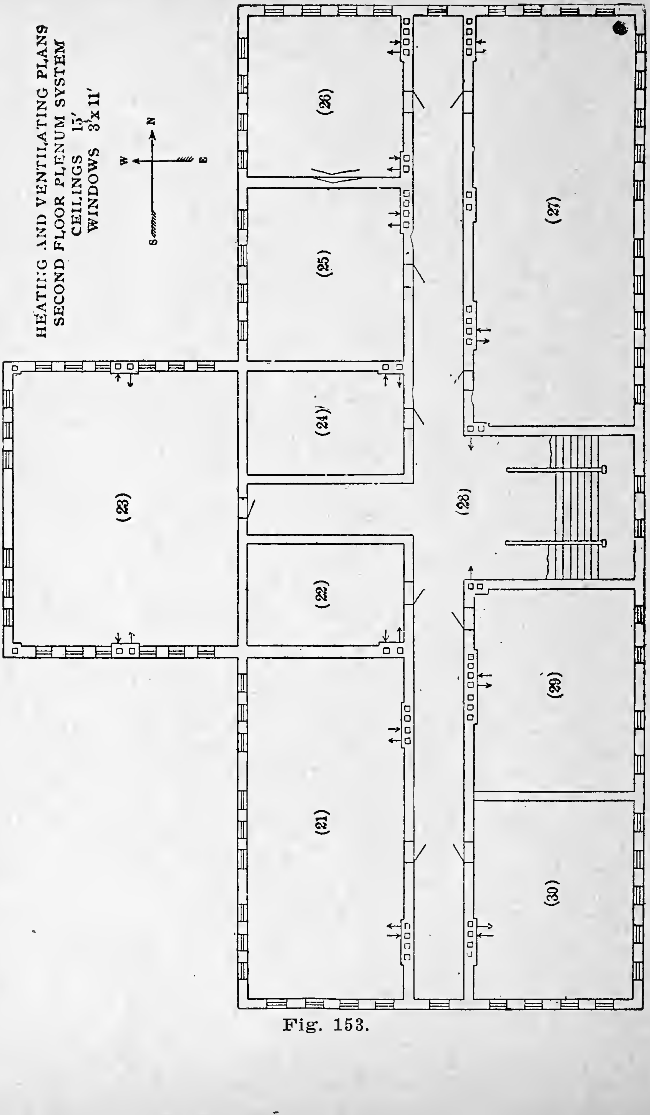
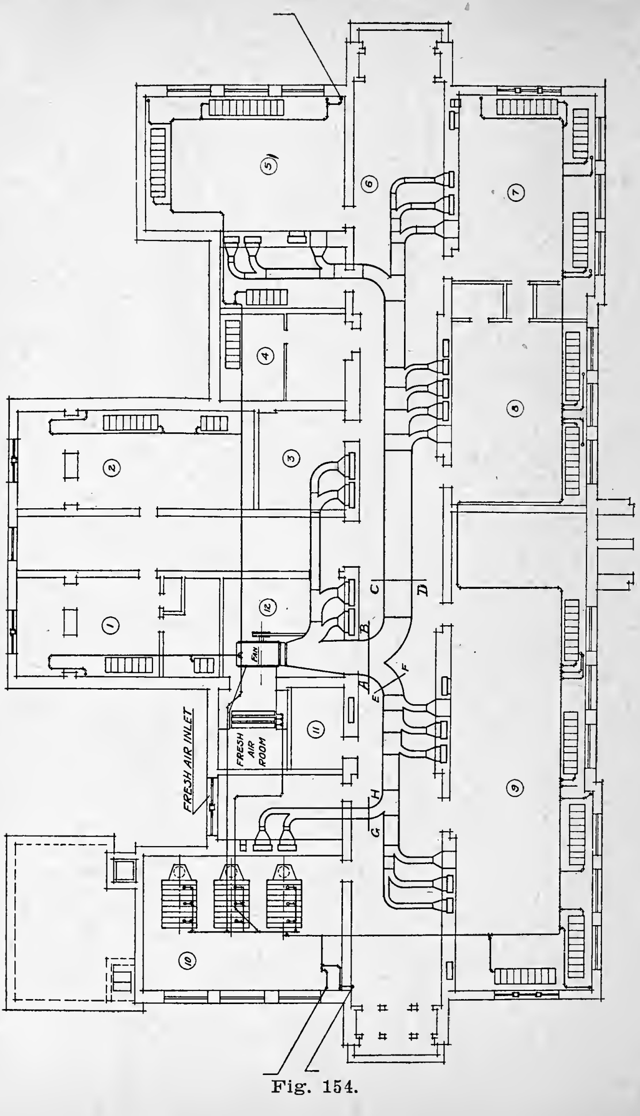

HEATING AND VENTILATING PLANS
SECOND FLOOR PLENUM SYSTEM
CEILINGS 15'
WINDOWS 3'x 11'
N
W
E
S
(21)
(22)
(23)
(24)
(25)
(26)
(27)
(28)
(29)
(30)

Fig. 153.

156. Application of Split System to School Building:— Figs. 154, 155 and 156 show the plans of a school building heated by direct-radiation and ventilated by fan-coil system. These are included especially to show the arrangements of the ducts and indirect apparatus on the basement plan. Some of the principal points in the design of the indirect section of this plant are as follows: Air moved by the fan per minute, 30000 cu. ft. against a static pressure of ¾ in. of water; fan, at a tip speed of 2700 f. p. m., requires 9 horse-power; motor horse-power, 12; 50″ vento coils, 3 stacks deep, 5″ centers, arranged in two tiers of 16 sections each making a total of 96 sections; air warmed from —10° to 80°; duct at *A B*, 15 sq. ft., velocity 1600 f. p. m.; at *C D*, 9 sq. ft., velocity 1500 f. p. m.; at *E F*, 7 sq. ft., velocity 1550 f. p. m., and at *G H*, 2.5 sq. ft., velocity 1150 f. p. m.; fresh air inlet grill 48 sq. ft., covered by ½″ mesh wire screen; individual exhaust ventilation for toilets and showers; automatic regulation on all direct radiation in all class rooms and on two of the three complete stacks in the vento heaters. To supply the coils and the direct radiation required three cast iron sectional boilers, each having a rated capacity of 8350 sq. ft. of direct steam radiation.



FRESH AIR INLET
FRESH AIR ROOM
FAN
1
2
3
4
5
6
7
8
9
10
11
12
A
B
C
D
E
F
G
H

Fig. 154.

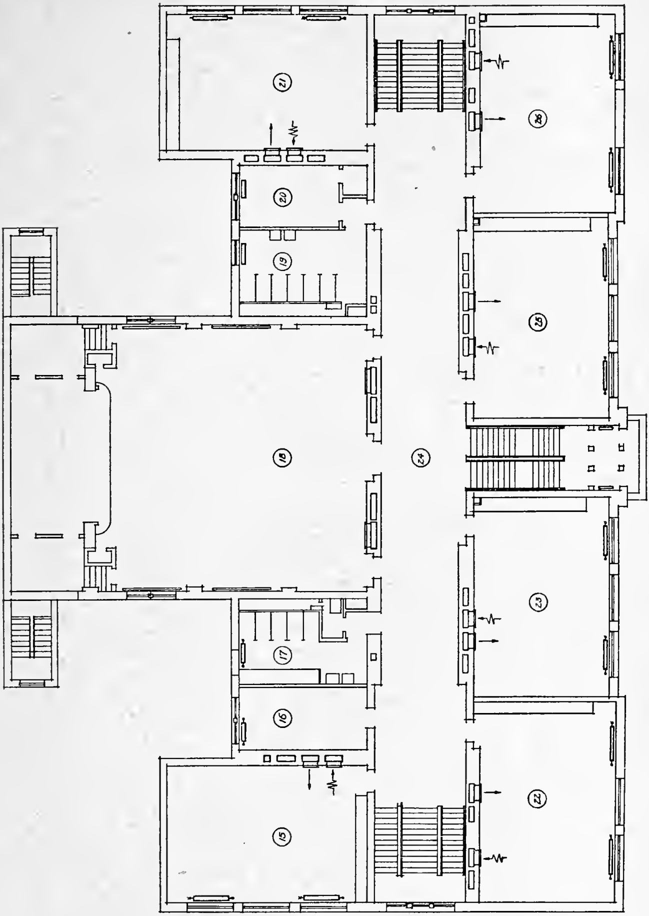


Fig. 155.

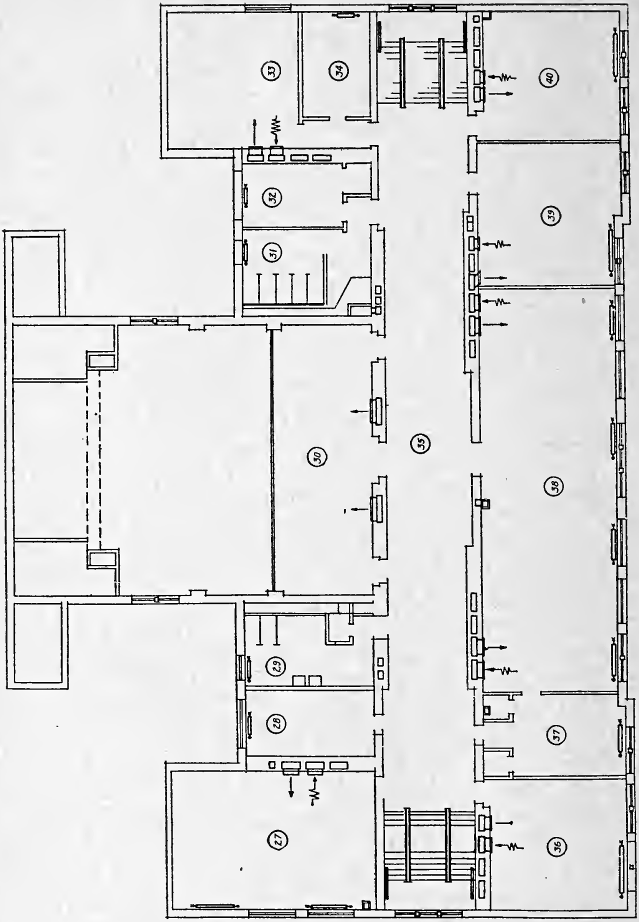


Fig. 156.

CHAPTER XIII.

DISTRICT HEATING OR CENTRALIZED HOT WATER AND STEAM HEATING.

GENERAL.

157. Heating Residences and Business Blocks from a central station is a method that is being employed in many cities and towns throughout the country. The centralization of the heat supply for any district in one large unit has an advantage over a number of smaller units in being able to burn the fuel more economically, and in being able to reduce labor costs. It has also the advantage, when in connection with any power plant, of saving the heat which would otherwise go to waste in the exhaust steam and stack gases, by turning it into the heating system. The many electric lighting and pumping stations around the country give large opportunity in this regard. Since the average steam power plant is very wasteful in these two particulars, any saving that might be brought about should certainly be sought for. On the other hand, however, a plant of this kind is at a disadvantage in that it necessitates transmitting the heating medium through a system of conduits, which generally is a wasteful process. The failure of many of the pioneer plants has cast suspicion upon all such enterprises as paying investments, but the successful operation of many others shows the possibilities, where care is exercised in their design and operation.

158. Important Considerations in Central Station Heating:—In any central heating system, the following considerrations will go far toward the success or the failure of the enterprise:

First.—There should be a demand for the heat.

Second.—The plant should be near to the territory heated.

Third.—There should be good coal and water facilities at the plant.

Fourth.—The quality of all the materials and the installation of the same, especially in the conduit concerning in-

sulation, expansion and contraction, and durability, are points of unusual importance.

Fifth.—The plant must be operated upon an economical basis, the same as is true of other plants.

Sixth.—The *load-factor* of the plant should be high. This is one of the most important points to be considered in combined heating and power work. The greater the proportion of hours each piece of apparatus is in operation, to the total number of hours that the plant is run, the greater the plant efficiency. The *ideal* load-factor requires that all of the apparatus be run at full load all the time.

The average conduit radiates a great deal of heat, hence, the nearer the plant to the heated district the greater the economy of the system. Likewise a location near a railroad minimizes fuel costs, and good water, with the possibility of saving the water of condensation from the steam, assists in increasing the economy of the plant. It is to be expected that even a well designed plant, unless safeguarded against ills as above suggested, would soon succumb to inevitable failure.

Two types of centralized heating plants are in use, *hot water* and *steam*. Each will be discussed separately. In the discussion of either system, certain definite conditions will have to be met. First of all, there should be a demand in that certain locality for such a heating system, before the plant can be considered a safe investment. To create a demand requires good representatives and a first-class residence or business district. When this demand is obtained the plan of the probable district to be heated will first be platted and then the heating plant will be located. In many cases the heating plant will be an added feature to an already established lighting or power plant and its location will be more or less a predetermined thing.

In addition to these material and financial features just mentioned, one must consider the legal phases that always come up at such a time. These relate chiefly to the franchise requirements that must be met before occupying the streets with conduit lines, etc. All of these considerations are a part of the one general scheme.

159. The Scope of the Work in central station heating may be had from the following outline:

- Central Station Heating
  - Hot Water Heating by use of..............
    - Exhaust steam heaters
    - Live steam heaters
    - Heating boilers
    - Economizers
    - Injectors or
    - Com-minglers
  - Steam Heating........
    - Exhaust steam
    - Live steam

In the *hot water system* the return water at a lowered temperature enters the power plant, is passed through one or more pieces of apparatus carrying live or exhaust steam, or flue gases, and is raised in temperature again to that in the outgoing main. From the above, a number of combinations of reheating can be had. Any or all of the units may be put in one plant and the piping system so installed that the water will pass through any single unit and out into the main; or, the water may be split and passed through units in series. All of these combinations are possible, but not practicable. In most plants, two or three combinations only are provided. In the existing plants the order of preference seems to be, exhaust steam reheaters, economizers, heating boilers, injectors or com-minglers, and live steam heaters.

All of the above pieces of reheating apparatus operate by the transmission of heat through metal surfaces, such as brass, steel or cast iron tubes, excepting the com-mingler, this being simply a barometric condenser in which the exhaust steam is condensed by the injection water from the return main, the mixture being drawn directly into the pumps.

The objection to the tube transmission is the lime, mud and oil deposit on the tube surfaces, thus reducing the rate of transmission and requiring frequent cleaning. The objections to the com-minglers are, first, that the pump must draw hot water from the condenser and second, that a certain amount of the oil passes into the heating line. With perfected apparatus for removing the oil, the com-mingler will no doubt supersede, to a large degree, the tube reheaters in hot water heating.

In the *steam system* the proposition is very much simplified. The exhaust steam passes through one or more oil separating devices and is then piped directly to the header leading to the outgoing main. Occasionally a connection is made from this line to a condenser, such that the steam, when not used in the heating system, may be run directly to the condenser. These pipe lines, of course, are all properly valved so that the current of steam may easily be deflected one way or the other. In addition to this exhaust steam supply, live steam is provided from the boiler and enters the header through a pressure reducing valve. In any case when the exhaust steam is insufficient the supply may be kept constant by automatic regulation on the reducing valve.

In selecting between hot water and steam systems the preference of the engineer is very largely the controlling factor. The preference of the engineer, however, should be formed from facts and conditions surrounding the plant, and should not come from mere prejudice. The following points are some of the important ones to be considered:

First cost of plant installed.—This is very much in favor of the steam system in all features of the power plant equipment, the relative costs of the conduit and the outside work being very much the same in the two systems.

Cost of operation.—This is in favor of the hot water system because of the fact that the steam from the engines may be condensed at or below atmospheric pressure, while the exhausts from the engines in the steam systems must be carried from five to fifteen pounds gage, which naturally throws a heavy back pressure upon the engine piston.

Pressure in circulating mains.—This is in favor of the steam system. The pressure in any steam radiator will be only a few pounds above atmosphere, while in a hot water system, connected to high buildings, the pressure on the first floor radiators near the level of the mains becomes very excessive. The elevation of the highest radiator in the circuit, therefore, is one of the determining factors.

Regulation.—It is easier to regulate the hot water system without the use of the automatic thermostatic control, since the temperature of the water is maintained according to a local schedule which fits all degrees of outside temperature.

When automatic control is applied, this advantage is not so marked.

Returning the water to the power plant.—In most steam plants the water of condensation is passed through indirect heaters to remove as much of the remaining heat as possible and is then run to the sewer. This procedure incurs a considerable loss, especially in cold weather when the feed water at the power plant is heated from low temperatures. This point is in favor of the hot water system.

Estimating charges for heat.—This is in favor of the steam system since, by meter measurement, a company is able to apportion the charges intelligently. The flat rate charged for water heating and for some steam heating is in many cases a decided loss to the company.

160. Conduits:—In installing conduits for either hot water or steam systems the selection should be made after determining, first, its efficiency as a heat insulator; second, its initial cost; third, its durability. Other points that must be accounted for as being very essential are: the supporting, anchoring, grading and draining of the mains; provision for expansion and contraction of the mains; arrangements for taking off service lines at points where there is little movement of the mains; and the draining of the conduit.

Some conduits may be installed at very little cost and yet may be very expensive propositions, because of their inability to protect from heat losses; while, on the other hand, some of the most expensive installations save their first cost in a couple of years' service. Many different kinds of insulating materials are used in conduit work such as magnesia, asbestos, hair felt, wool felt, mineral wool and air cell. Each of these materials has certain advantages and under certain conditions would be preferred. It is not the real purpose here to discuss the merits of the various insulators, because the quality of the workmanship in the conduit enters into the final result so largely. The different ways that pipes may be supported and insulated in outside service will be given, with general suggestions only. Figs. 157 and 158 show a few of the many methods in common use. A very simple conduit is shown at *A*. This is built up of wood sections fitted end to end, covered with tarred paper to prevent surface water leaking in and bound with straps. The pipe either is a loose fit to the bore and rests upon the inner sur-

face, or is supported on metal stools, driven into the wood or merely resting upon it. Stools hold the pipe concentric with the inner bore of the log. With much end movement of the pipe, from expansion and contraction, stools should not be used unless they are loose and have a wide surface contact with the wood. A metal lining with the pipe resting directly upon it is considered good. The conduit is laid to a good straight run in a gravel bed and usually over a small tile drain to carry off the surface water, excepting as this drain is not necessary in sections where there is good gravel drainage. The insulation in *A* is only fair. The air space around the pipe, however, is to be commended. *B* is an improvement over *A* and is built up of boards notched at the edges to fit together. The materials used, from the outside to the center, are noted on the sketch beginning with the top and reading down. This covering is in general use and gives good satisfaction from every standpoint. *C* shows a good insulation and supports the pipe upon rollers at the center of a line of halved, vitrified tile. The lower half of the tile should be graded and the pipe then run upon the rollers, after which it may be covered with some prepared covering and the remaining space next the tile filled with asbestos, mineral wool or other like material. *D* shows the same adapted to basement work. Occasionally two pipes are run side by side, main and return, in which case large halved tiles may be used as in *E*, having large metal supports curved on the lower face to fit the tile. If these supports are not desired the same kind of straight tiles may be used with a tee tile inserted every 8 to 12 feet having the bell looking down as in *F*. In this bell is built a concrete setting with iron supports for the pipes which run on rollers. These rollers are sometimes pieces of pipes cut and reamed, but are better if they are cast with a curvature to fit the pipes to be supported. This form of conduit, when drained to good gravel, gives first-class service. *G*, *H* and *I* show box conduits with two or more thicknesses of ⅞ inch boards nailed together for the sides, top and bottom. The bottom of the conduit is first laid and the pipe is run. The sides are then set in place and the insulating material put in, after which the top is set and the whole filled in. *I* shows the best form of box, since with the air spaces this is a very good insulator. All wood boxes are very temporary,

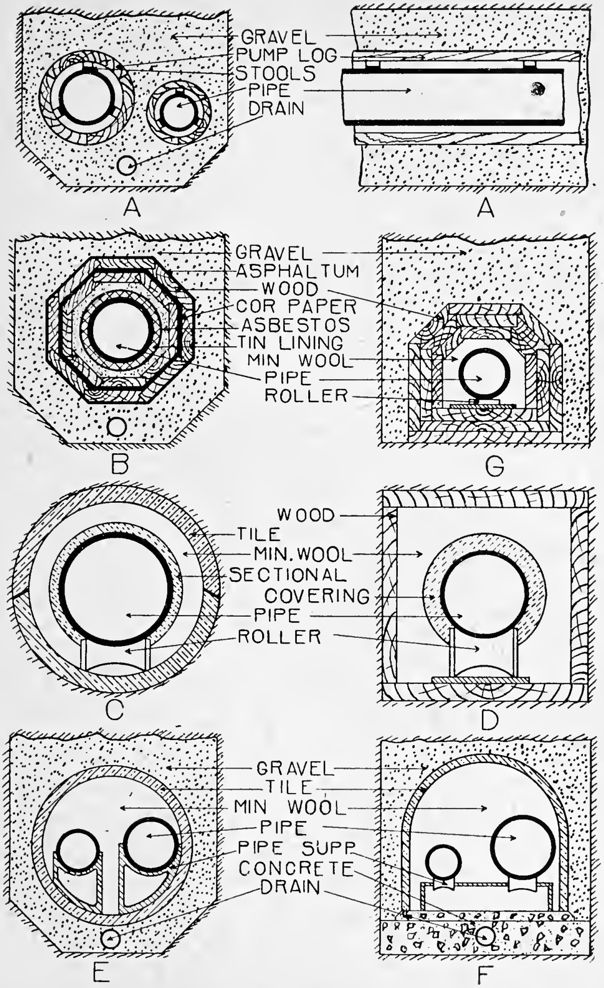

GRAVEL
PUMP LOG
STOOLS
PIPE
DRAIN
A
A
GRAVEL
ASPHALTUM
WOOD
COR PAPER
ASBESTOS
TIN LINING
MIN WOOL
PIPE
ROLLER
B
G
WOOD
TILE
MIN. WOOL
SECTIONAL
COVERING
PIPE
ROLLER
C
D
GRAVEL
TILE
MIN WOOL
PIPE
PIPE SUPP
CONCRETE
DRAIN
E
F

Fig. 157.

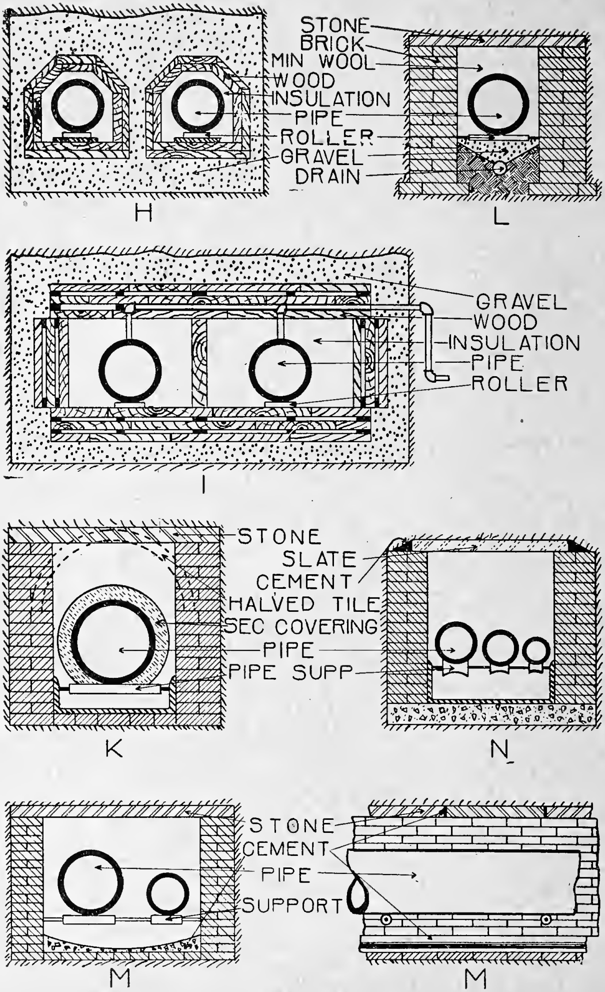

STONE
BRICK
MIN WOOL
WOOD
INSULATION
PIPE
ROLLER
GRAVEL
DRAIN
H
L
GRAVEL
WOOD
INSULATION
PIPE
ROLLER
I
STONE
SLATE
CEMENT
HALVED TILE
SEC COVERING
PIPE
PIPE SUPP
K
N
STONE
CEMENT
PIPE
SUPPORT
M
M

Fig. 158.

hence, brick and concrete are usually preferred. *K* is a conduit with 8 inch brick walls covered with flat stones or halved glazed tiles cemented to place to protect from surface leakage. The bottom of the conduit has supports built in every 8 to 12 feet, and between these points the conduit drains to the gravel. The usual rod and roller here serve as pipe supports. The pipe is covered with sectional covering and the rest of the space may or may not be filled with wool or chips, as desired. *L* shows the sectional covering omitted and the entire conduit filled with mineral wool, hair felt or asbestos. *M* has the supporting rod built into the sides of the conduit and has the bottom of the conduit bricked across and cemented to carry the leaks and drainage to some distant point. *N* shows a concrete bottom with brick sides, having the pipe supported upon cast iron standards. The latest conduit has concrete slabs for bottom and sides and has a reinforced concrete slab top. This comes as near being permanent as any, is reasonable in price, and when the interior is filled with good non-conducting material, or when the pipe is covered with a good sectional covering, it gives fairly high efficiency.

All conduit pipes should be run as nearly uniform in grade as possible to avoid the formation of air and water pockets. Any unusual elevation in any part of the main may require an air vent being placed at the uppermost point of the curve, otherwise air may collect in such quantities as to retard circulation. All low points in the steam lines must be drained to traps.

The *heat loss from conduits* is an item of considerable importance. A good quality of materials and insulation will probably reduce this loss as low as 20 to 25 per cent. of the amount lost from the bare pipe. To show the method of analysis and to obtain an estimate of the average conduit losses, the following application will be made to a supposed two-pipe hot water system. The loss of heat in B. t. u. per lineal foot from any pipe per hour may be taken from the equation

$$H_c = KCA\ (t - t') \tag{90}$$

where K = rate of transmission for uncovered pipes, C = 100 per cent. — efficiency of the insulation, A = area of pipe surface per lineal foot of pipe, t = average temperature on the

inside of the pipe and $t' =$ average temperature on the outside of the conduit.

APPLICATION.—Having given a system of conduit pipes (two pipes in one conduit) with sizes and lengths as stated in the first and second columns of Table XXXVI, what is the probable heat loss in B. t. u. per hour on a winter day and what is the radiation equivalent in a hot water system carrying water at an average temperature of 170 degrees?

TABLE XXXVI.

Pipe size inches	Total lineal feet of main and return	Surface per foot of length A	B. t. u. per hr. per lineal foot H_c	Equivalent no. of sq. ft. of H. W. Rad.
2	5000	.62	48.8	1435
3	2000	.91	71.6	842
4	3000	1.06	83.4	1472
6	3000	1.73	137.1	2420
8	2000	2.26	177.9	2093
10	2000	2.83	221.9	2611
12	2000	3.33	262.0	3082
14	1000	4.00	314.8	1852
Totals. B. t. u. lost per hour 2687100				15807

If $K = 2.25$, $C = 100 - 75 = 25$ per cent., $t = 175$ and $t' = 35$, we have for a 2-inch pipe, $H_c = 2.25 \times .25 \times .62 \times 140 = 48.8$, which for 5000 lineal feet $= 244000$ B. t. u., and for the entire system, 2687100 B. t. u. If each square foot of hot water radiation gives off 170 B. t. u. per hour then the radiation equivalent for the 2-inch pipe is $244000 \div 170 = 1435$ square feet. Similarly work out for each pipe size and obtain the values given in the last column of the table. This conduit loss is sufficient to heat 15807 square feet of radiation in the district. In terms of the coal pile it approximates 350 pounds per hour. Now assuming the 14 inch main to supply the entire district at a velocity of 6 feet per second we have approximately 162000 square feet of H. W. surface on the line. From this the line loss is $15807 \div 162000 = 9.1$ per cent. It should be remembered that the above assumes the plant working under a heavy load, when the per cent.

of line loss is a minimum. This loss remains fairly constant while the heat utilized in the district fluctuates greatly. In mild weather, therefore, the per cent. of line loss to the total heat transmitted is much greater.

161. Layout of Street Mains and Conduits:—No definite information can be given concerning the layout of street mains, because the requirements of each district would call for independent consideration. The following general suggestions, however, can be noted as applying to any hot water or steam system:

Streets to be used.—Avoid the principal streets in the city, especially those that are paved; alleys are preferred because of the minimum cost of installation and repairs.

Cutting of the mains.—Do not cut the main trunk line for branches more often than is necessary. Provide occasional by-pass lines between the main branches at the most important points in the system, so that, if repairs are being made on any one line, the circulation beyond that point may be handled through the by-pass. Such by-pass lines should be valved and used only in case of emergency.

Offsets and expansion joints.—Offsets in the lines hinder the free movement of the water and add friction head to the pumps; hence, in water systems, the number should be reduced to a minimum. Long radius bends at the corners reduce this friction. Offsets are especially valuable to take up the expansion and contraction of the piping without the aid of expansion joints. This is illustrated in Fig. 159, where anchors are placed at A, and the gradual bending of the pipes at each corner makes the necessary allowance. The expansion in wrought iron is about .00008 inch per foot per degree rise in temperature; hence in a hot water main the lineal expansion between 0° and 212° is .017 inch per foot of length or 1.7 inches for each 100 feet of straight pipe. In hot water heating systems the temperature of this pipe would never be less than 50°, which would cause an expansion from hot to cold of only .013 inch per foot, or 1.3 inches for each 100 feet of straight pipe. In steam systems the pipe temperature may vary anywhere from 50° to 300°, making a lineal expansion of .02 inch per foot of length or 2 inches for each 100 feet of straight pipe. As here shown the

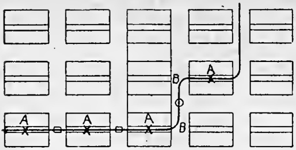


Fig. 159.

movement from the anchor at *A* toward *B* may be absorbed by the swinging of the pipe about *O*. *B.B.* should therefore be as long as possible to avoid unduly straining the pipe at the joints. Allowing a maximum movement of 6 inches for each expansion joint, the anchors would be spaced 500 and 300 feet center to center respectively, for hot water and steam mains. These figures would seldom be exceeded, and in some cases would be reduced, the spacing depending upon the type of expansion joint used. Ordinarily, 400 feet spacing can be recommended for hot water and 300 feet for steam. If the city layout meets this value fairly well, then the expansion joints and anchors may be made to alternate with each other, one each to every city block.

A few of the expansion joints in common use are shown in Fig. 160. *A* is the old slip and packed joint. This joint causes very little trouble except that it needs repacking frequently. It is very effective when properly cared for. The slip joint should have bronze bearings on both the outside of the plug and the lining of the sleeve. The ends of the plug and sleeve may be screwed for small pipes, or flanged for large ones. *B* shows an improved type of slip joint, having a roller bearing upon a plate in the bottom of the conduit, and plugs bearing against metal plates along the sides of the conduit to keep it in line. *C* and D show other slip joints very similar to *A* and *B*. *C* has one ball and socket end to adjust the joint to slight changes in the run of the pipe, and *D* has two packings enclosing the plug to give it rigidity. The drainage in each case is taken off at the bottom of the casting. *E* has two large flexible disks fastened to the ends of the pipe and separated from each other by an annular ring casting. These disks are frequently corrugated, are usually of copper and are very large in diameter so that the pipe has considerable movement without endangering the metal in the disks. *F* has a corrugated copper tube fastened at the ends to the pipe flanges. This is protected from excessive internal pressure by a straight tube having a sliding fit to the inside of the flanges, thus allowing for end movement. *G* is very similar to *E*. It has,

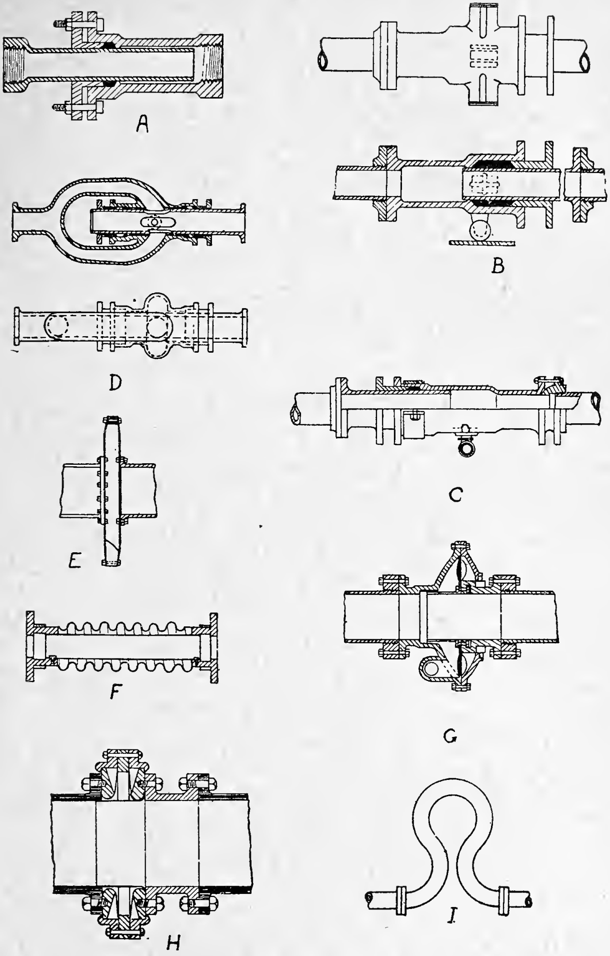


Fig. 160.

however, only one copper disk. This disk is enclosed in a cast iron casement, one side of which is open to the atmosphere, the other side having the same pressure as within the pipe. *H* is very similar to *E*, having two copper diaphragms to take up the movement. These diaphragms flex over rings with curved edges and are thus protected somewhat against failure. *I* shows a copper U tube which is sometimes used. This is set in a horizontal position and the expansion and contraction is absorbed by bending the loop. In all these joints those which depend upon the bending of the metal require little attention except where complete rupture occurs. In old plants, however, the rupturing of these diaphragms is of frequent occurrence. The packed joint requires attention for packing several times in the year, but very seldom causes trouble other than this.

Anchors.—In any long run of pipe, where the expansion and contraction of the pipe causes it to shift its position very much, it is necessary to anchor the pipe at intervals so as to compel the movement toward certain desired points. The anchor is sometimes combined with the expansion joint, in which case the conduit work is simplified (See Fig. 161).

Service pipes to residences are preferably taken off at

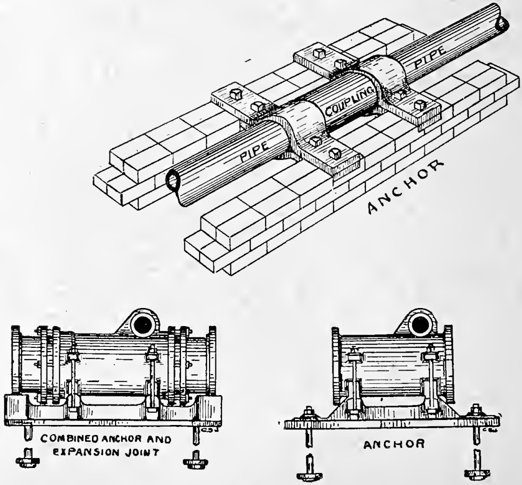


Fig. 161.

or near the anchors. All condensation drains in steam mains are likewise taken off at such points.

Valves.—All valves on water systems should be straightway gate valves. Valves on steam systems should be gate valves on lines carrying condensation, and may be renewable seat globe valves on the steam lines. Valves should be placed on the main trunk at the power plant, on all the principal branch mains as they leave the main trunk, on all by-pass lines, on all the service mains to the houses, and at such important points along the mains as will enable certain portions of the heating district to be shut off for repairs without cutting out the entire district.

Manholes.—Manholes are placed at important points along the line to enclose expansion joints and valves. These manholes are built of brick or concrete and covered with iron plates, flag stones, slate or reinforced concrete slabs. Care must be exercised to drain these points well and to have the covering strong enough to sustain the superimposed loads.

162. Typical Design for Consideration:—In discussing district heating, each important part of the design work will be made as general as possible and will be closed by an

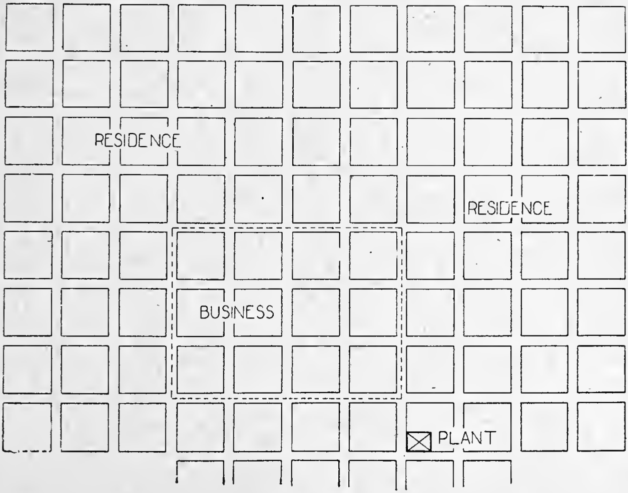


Fig. 162.

application to the following concrete example which refers to a certain portion of an imaginary city (Fig. 162) as available territory. A city water supply and lighting plant is located as shown, with lighting and power units aggregating 475 *K. W.*, city water supply pumps aggregating 3000000 gallons maximum capacity, and smaller pumps requiring approximately 15 per cent. of the amount of steam used by the larger lighting units. It is desired to re-design this plant and to add a district heating system to it; the same to have all the latest methods of operation and to be of such a size as to be economically handled. Fig. 169 shows the essential details of the finished plant.

163. Electrical Output and Exhaust Steam Available for Heating Purposes from the Power Units:—In the operation of such a plant, one of the principal assets is the amount of exhaust steam available for heating purposes. The amount may be found for any time of the day or night by constructing a power chart as in Fig. 163, and a steam consumption chart as in Fig. 164. Referring to Fig. 163, the values here

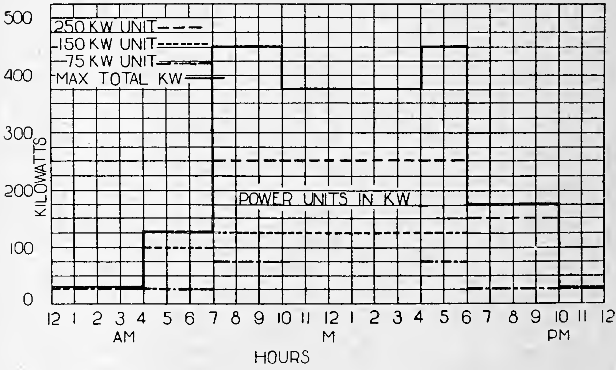


Fig. 163.

given are assumed, for illustration, to be those recorded at the switchboard of the typical plant on a day when heavy service is required. The curves show that the 75 *K. W.* unit runs from 12 P. M. to 7 A. M. and from 6 P. M. to 12 P. M. with an output of 25 *K. W.* It also runs from 7 A. M. to 10 A. M. and from 4 P. M. to 6 P. M. under full load. The 150 *K. W.*

unit runs from 4 A. M. to 7 A. M. with an output of 100 *K. W.* and then increases to 125 *K. W.* for the entire time until 6 P. M. when it is shut down. The 250 *K. W.* unit is started up at 7 A. M. and runs until 6 P. M. under full load, when the load drops off to 150 *K. W.* and continues until 10 P. M. when the unit is shut down, leaving only the 75 *K. W.* unit running. The heavy solid line shows all the power curves superimposed one upon the other. Having given the *K. W.* output, the general equation for determining the horse-power of the engines is

$$I.\ H.\ P. = \frac{K.\ W. \times 1000}{746 \times E \times E'} \qquad (91)$$

where E and E' are the efficiencies of the generator and engine respectively. If we assume the efficiency of the generator to be 90 per cent., and that of the engine to be 92 per cent., then Equation 91 becomes

$$I.\ H.\ P. = \frac{K.\ W. \times 1000}{746 \times .90 \times .92} = \text{approx. } 1.62\ K.\ W. \qquad (92)$$

Assuming that the 250 *K. W.* unit consumes 24 pounds, the 150 *K. W.* unit 32 pounds, and the 75 *K. W.* unit 32 pounds of steam per *I. H. P.* hour respectively, when running under

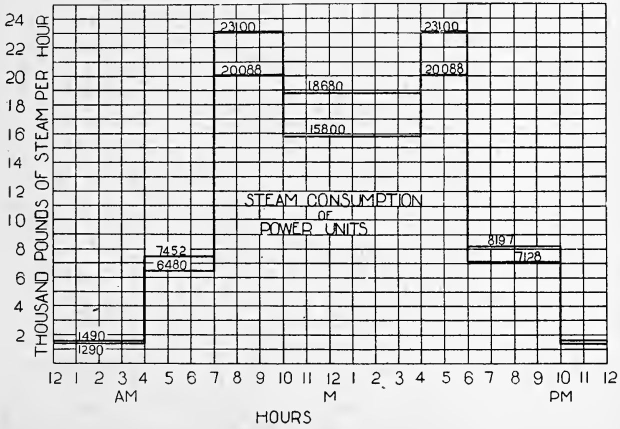


Fig. 164.

normal loads, the total steam consumed in the three units at any time is shown by the lower curve in Fig. 164. The upper curve shows the 15 per cent. added allowance for smaller units not included in the above list. The values assumed for efficiencies and the values for steam consumption are reasonable and may be used if a more exact figure is not to be had.

It will be seen that the maximum steam consumption in the generating units in the power plant is 23100 pounds per hour and the minimum is 1490 pounds per hour. These two amounts, together with the exhaust steam from the circulating pumps on the heating system, if a hot water system is installed, and that from the pumps in the city water supply, will determine the capacity of the exhaust steam heaters on the hot water supply and the capacity of the boilers or economizers to be used as heaters when the exhaust steam is deficient.

164. Amount of Heat Available for Heating Purposes in Exhaust Steam, Compared with That in Saturated Steam at the Pressure of the Exhaust:—To study the effect of exhaust steam upon heating problems and to determine, if possible, the theoretical amount of heat given off with the exhaust steam under various conditions of use, make several applications: first, to a simple high speed non-condensing engine using saturated steam; second, to a compound Corliss non-condensing engine using saturated steam; third, to the first application when superheated steam is used instead of saturated steam; and fourth, to a horizontal reciprocating steam pump. Assume the following safe conditions. Case one—boiler pressure 100 pounds gage; pressure of steam entering cylinder 97 pounds gage; quality of steam at cylinder 98 per cent.; steam consumption 34 pounds per indicated horse-power hour; one per cent. loss in radiation from cylinder; and exhaust pressure 2 pounds gage. Case two—boiler pressure 125 pounds gage; pressure at high pressure cylinder 122 pounds gage; quality of steam entering high pressure cylinder 98 per cent.; steam consumption 22 pounds per indicated horse-power hour; 2 per cent. loss in radiation from cylinders and receiver pipe, and exhaust pressure 2 pounds gage. Case three—same as case one with superheated steam at 150 degrees of superheat. Case four—as stated later.

The number of B. t. u. exhausted with the steam, in any case, is the total heat in the steam at admission, minus the heat radiated from the cylinder, minus the heat absorbed in actual work in the cylinder.

High speed engine. Case one.—Let r = heat of vaporization per pound of steam at the stated pressure, x = quality of the steam at cut-off, q = heat of the liquid in the steam per pound of steam, and W_s = pounds of steam per indicated horse-power hour. From this the total number of B. t. u. entering the cylinder per horse-power hour is

$$\text{Total B. t. u.} = W_s\ (xr + q) \tag{93}$$

From Table 4, $r = 881$, $x = .98$ and $q = 307$; then if $W_s = 34$, initial B. t. u. $= 34\ (.98 \times 881 + 307) = 39792.92$. Deducting the heat radiated from the cylinder we have $39792.92 \times .99 = 39395$ B. t. u. per horse-power left to do work. The B. t. u. absorbed in mechanical work (useful work + friction) in the cylinder per horse-power hour is $(33000 \times 60) \div 778 = 2545$ B. t. u. Subtracting this work loss we have $39395 - 2545 = 36850$ B. t. u. given up to the exhaust per horse-power hour. Comparing this value with the total heat in the same weight of saturated steam at 2 pounds gage, we have $100 \times 36850 \div (34 \times 1152.8) = 94$ per cent.

Compound Corliss engine. Case two.—With the same terms as above let $r = 869$, $x = .98$, $q = 322.8$, and $W_s = 22$, then the initial B. t. u. $= 22\ (.98 \times 869 + 322.8) = 25837$. Less 2 per cent. radiation loss $= 25837 \times .98 = 25321$ B. t. u. The loss absorbed in doing mechanical work in the cylinder per horse-power is, as before, 2545 B. t. u. Subtracting this we have $25321 - 2545 = 22776$ B. t. u. given up to the exhaust per horse-power hour. Comparing as before with saturated steam at 2 pounds gage, we have $100 \times 22776 \div (22 \times 1152.8) = 90$ per cent.

Case three.—Now suppose superheated steam be used in the first application, all other conditions being the same, the steam having 150 degrees of superheat, what difference will this make in the result? The total heat entering the cylinder now is the total heat of the saturated steam at the initial pressure plus the heat given to it in the superheater. Let c_p = specific heat of superheated steam and

t_d = the degrees of superheat, then the total heat of the superheated steam is

$$\text{Total B. t. u. (sup.)} = W_s\ (xr + q + c_p t_d) \qquad (94)$$

This for one horse-power of steam (34 pounds), if the specific heat of superheated steam is .54, will be 34 × .99 × (1188 + .54 × 150) = 42714.5 B. t. u. and the heat turned into the exhaust will be 42714.5 — 2545 = 40169.5 B. t. u. Comparing this with the heat in saturated steam at 2 pounds gage, we have 100 × 40169.5 ÷ (34 × 1152.8) = 102 per cent.

Case four.—Pump exhausts are sometimes led into the supply and used for heating purposes along with the engine exhausts. If such conditions be found, what is the heating value of such steam? Assume the live steam to enter the steam cylinder of the pump under the same pressure and quality as recorded for the high speed engine. The steam is cut off at about ⅞ of the stroke and expands to the end of the stroke. With this small expansion the absolute pressure at the end of the stroke will be approximately ⅞ × 112 = 98 pounds, and if enough heat is absorbed from the cylinder wall to bring the steam up to saturation at the release pressure, we will have a total heat above 32 degrees, in the exhaust steam per pound of steam at 98 pounds absolute, of 1186 B. t. u. Comparing this with a pound of saturated steam at 2 pounds gage, we have 100 × 1186 ÷ 1152.8 = 103 per cent. Under the conditions such as here stated with a high release pressure, a small expansion of steam in the cylinder and dry steam at the end of the stroke, it is possible to suddenly drop the pressure from pump release to a low pressure, say 2 pounds gage, and have all the steam brought to a state approaching superheat. It is not likely, however, that the steam is dry at the end of the stroke in any pump exhaust, because the heat lost in radiation and in doing work in the slow moving pump would be such as to have a considerable amount of entrained water with the steam, thus lowering the quality of the steam. These above conditions are extreme and are not obtained in practice.

From cases one and two it would appear that the *greatest* amount of heat that can be expected from *engine* exhausts, for use in heating systems at or near the pressure of the atmosphere, is 90 to 94 per cent. of that of

saturated steam at the same pressure. The percentage will, in most cases, drop much below this value. All things considered, *exhaust steam having 80 to 85 per cent. of the value of saturated steam at the same pressure is probably the safest rating when calculating the amount of radiation which can be supplied by the engines.* In many cases no doubt this could be exceeded, but it is always best to take a safe value. On the other hand, *when figuring the amount of condenser tube surface or reheater tube surface to condense the steam, it would be best to take exhaust steam at 100 per cent. quality,* since this would be working toward the side of safety.

In plants where the exhaust steam is used for heating purposes and where the amount supplied by direct acting steam pumps is large compared with that supplied by the power units, it is possible to have the quality of the exhausts anywhere between 800 and 1000 B. t. u. per pound of exhaust. It should be understood that saturated steam at *any* stated pressure always has the same number of B. t. u. in it, no matter whether it is taken directly from the boiler, or from the engine exhaust. A pound of the mixture of steam and entrained water, taken from engine exhausts, should not be considered as a pound of steam. If we are speaking of a pound of exhaust steam without the entrained water as compared with a pound of saturated steam at the same pressure, they are the same, but a pound of engine exhaust or mixture is a different thing.

HOT WATER SYSTEMS.

165. Four General Classifications of hot water heating may be found in current work, two applying to the conduit piping system and two to the power plant piping system. The first, known as the *one-pipe complete circuit system,* is shown in Fig. 165. It will be noticed that the water leaves the power plant and makes a complete circuit of the district, as *A, B, C, D, E, F, G,* through a single pipe of uniform diameter. From this main are taken branch mains and leads to the various houses, as *a, b, c* and *d, e,* each one returning to the principal main after having made its own minor circuit. The second is known as the *two-pipe high pressure system,* in which two main pipes of like diameter laid side by side in the same conduit, radiate from the power plant to the farthest point on the line reducing in size at certain points to suit the capacity of that part of the district

served. This system is represented by Fig. 166. In the one-pipe system the circulation in the various residences is maintained, in part, by what is known as the *shunt* system, and in part, by the natural gravity circulation. The circulation in the two-pipe system is maintained by a high differential pressure between the main and the return at the same point of the conduit. The force producing movement of the water in the shunt system is, therefore, very much less than in the two-pipe system. As a consequence,

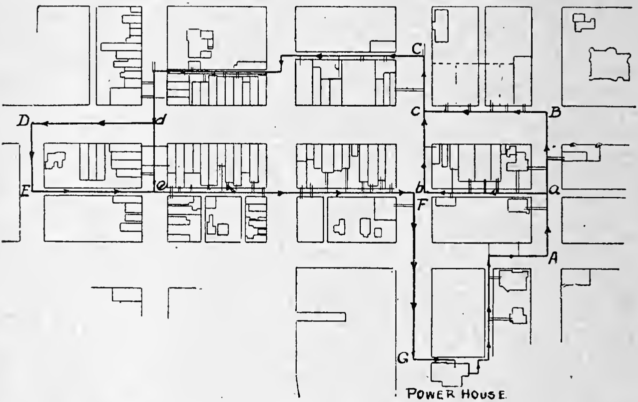


Fig. 165.

the one-pipe system has a lower velocity of the water in the houses and larger service pipes than the two-pipe system.

In many cases it is desired to connect central heating mains to the low pressure hot water systems in private plants. Such connections may easily be made with either one of the two systems by installing some minor pieces of apparatus for controlling the supply.

The third and fourth classifications, the *open* and *closed* systems, have about the same meaning as when applied to gravity work in isolated plants. The first is open to the atmosphere at some point along the circulating system, usually at the expansion tank which is placed on the return line just before the circulating pumps. The closed system presupposes some form of regulation for controlling excessive or deficient pressures without the aid of an expansion tank. In such cases pumps with automatic control may be used for taking care of the reserve supply of water. In the

open system the exhaust steam may be injected directly into the return circulating water by the use of an open heater or a com-mingler. The open heater and com-mingler cannot be used on the pressure side of the pumps. Surface condensers or reheaters, heating boilers and economizers may be used on either open or closed systems.

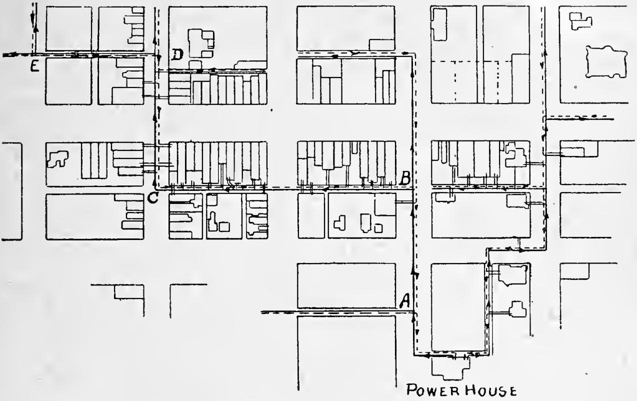


Fig. 166.

166. Amount of Water Needed per Hour as a Heating Medium:—All calculations must necessarily begin with the heat lost at the residence. Referring to the living room, Table XX, the heat loss is 15267 B. t. u. per hour, requiring 91 square feet of hot water heating surface to heat the room. Let the circulating water have the following temperatures: leaving the power plant 180°, entering the radiator 177°, leaving the radiator 157°, and entering the power plant 155°. According to these figures, which may be considered fair average values, the water gives off to the radiator 20 B. t. u. per pound or 166.6 B. t. u. per gallon, thus requiring $15267 \div 166.6 = 91$ gallons of water per hour to maintain the room at a temperature of 70°. From this a safe estimate may be given for design, *allow one gallon of water per hour for each square foot of hot water heating surface in the district.* In a plant operating under high efficiency this may be reduced to 6 pounds per square foot per hour. It is very certain that some plants are designed to supply less than one gallon, but in such cases it requires a higher tem-

perature of the circulating water and allows little chance for future expansion of the plant. A drop of 20 degrees, i. e., 20 B. t. u. heat loss per pound of water passing through the radiator, is probably the most satisfactory basis. All things considered, the above italicised statement will satisfy every condition. (See Art. 173). Having the total number of square feet of radiation in the district, the total amount of water circulated through the mains per hour can be obtained, after which the size of the pumps in the power plant may be estimated.

167. Radiation in the District:—The amount of radiation that may be installed in the district is problematical. In an average residence or business district the following figures may easily be realized: *business block, 9000 square feet; residence block, 4500 square feet.* In certain locations these figures may be exceeded and in others they may be reduced. Where the needs of the district are thoroughly understood a more careful estimate can easily be made. It is always well to make the first estimate a safe one and any possible increase above this figure could be taken care of as in Art. 166. Referring to Fig. 162, an estimate of the amount of radiation that may be expected in this typical case, if we assume ten business blocks and twenty-one residence blocks, is 184500 square feet. This will call for the circulation of 184500 gallons of water per hour.

168. Future Increase in Radiation:—From the temperatures given in Art. 166, it will be seen that each pound of water takes on 25 B. t. u. at the power plant and that there is a possible increase of 212 — 180 = 32 B. t. u. per pound that *may* be given to it, thus increasing the capacity of the system approximately 125 per cent. It would not be safe to count on such an increase in the average plant because of a defective layout in the piping system or because of a low efficiency in some of the pumps or other apparatus in the plant. If, however, a plant is installed according to the above figures, the capacity may be quite materially increased by increasing the temperature of the outgoing water at the plant to 212°.

169. The Pressure of the Water in the Mains:—The elevation above the plant at which a central station can supply radiation is limited. Water at 180° will weigh 60.55 pounds

per cubic foot, and the pressure caused by an elevation of 1 foot is .42 pound per square inch. From this the static pressure at the power plant, due to a hydraulic head of 100 feet, is 42 pounds per square inch. This value should not be exceeded, and generally, because of the influence it has on the machines and pipes toward producing leaks or complete ruptures, a less head than this is desirable. A static pressure of 42 pounds may be expected to produce, in a well designed plant, an outflow pressure of 65 to 75 pounds per square inch and a return pressure of 15 to 20 pounds per square inch, when working under fairly heavy service. In any case where the mains are too small to supply the radiation in the system properly, we may expect the value given for the outflow to increase and that for the return to decrease. A safe set of conditions to follow is: head, in feet, 60; static pressure, in pounds per square inch, 25; outgoing pressure at the pumps, in pounds per square inch, 50; return pressure at the pumps, in pounds per square inch, 5. This differential pressure of 45 pounds is caused by the friction losses in the piping system, pumps and heaters. *Long pipe systems,* as these are called, have much greater friction losses in the long runs of piping than in the ells, tees, valves, etc., hence, the friction head of the pipes is all that is usually considered. Where the minor losses are thought to be large, they may be accounted for by adding to the pipe loss a certain percentage of itself, say 10 to 20 per cent. *Pump power is figured from the differential pressure.*

The maximum and minimum pressures in the system are due to two causes: first, the static head, and second, the frictional resistances. These extremes of pressure are approximately—*static head plus (or minus) one-half the frictional resistances.* To obtain the frictional resistances, Chezy's Equation 95, is recommended. See Merriman's "A Treatise on Hydraulics," Arts. 86 and 100, and Church's "Mechanics of Engineering," Art. 519.

$$h_f = \frac{4\phi l}{d} \times \frac{v^2}{2g} \qquad (95)$$

where h_f = feet of head lost in friction, ϕ = friction factor (synonymous with coefficient of friction. For clean cast iron pipes with a velocity of 5 to 6 feet per second this has been found to vary from .0065 to .0048 for diameters between 3 and 15 inches respectively. .005 is suggested as

a safe average value to use), l = length of pipe in feet, v = velocity of water in feet per second, d = diameter of pipe in feet and $2g = 64.4$.

APPLICATION.—In Fig. 166, let it be desired to find the differential pressure at the pumps due to the friction losses in the line A, B, C, D, E. The lengths of the various parts are: power plant to A, 200 feet; A to B, 500 feet; B to C, 1500 feet; C to D, 1500 feet; and D to E, 500 feet. Assume, for illustration, that the total radiation in square feet beyond each of these points is: power plant, 125000; A, 85000; B, 50000; C, 28000; and D, 12000. This requires 125000, 85000, 50000, 28000 and 12000 gallons of water per hour, or 4.74, 3.27, 1.75, 1 and .44 cubic feet of water per second, respectively, passing these points. Now, if the velocities be roughly taken at 6 and 5 feet per second, (pipes near the power plant may be given somewhat higher velocities than those at some distance from the plant), the pipes will be 12, 10, 8, 6 and 4 inches diameter. In applying the equation to one part of the line we show the method employed for each. Take that part from the power plant to A. With $v = 6$

$$h_f = \frac{4 \times .005 \times 200 \times 36}{64.4 \times 1} = 2.2 \text{ feet.}$$

It should be noted here that Equation 95 refers to pipes where *all the water that enters at one end passes out the other.* This is not true in heating mains where a part of the water is drawn off at intermediate points. On the other hand, Merriman (Art. 99) explains that a water service main, where the water is *all taken off from intermedite tappings* and where the *velocity at the far end is zero,* causes only one-third of the friction given by the above equation. The case under consideration falls somewhere between these two extremes, the part next the power plant approaching the former and the last part of the line exactly meeting the conditions of the latter. Assuming the mean of these two conditions, which is probably very close to the actual, gives two-thirds of that found by the equation. Now since this is a double main system, i. e., main and return of the same size, the friction head for the two lines becomes 2.94 feet, from the power plant to A. In a similar way the other parts may be tried and the results from the entire line assembled in convenient form as in Table XXXVII.

TABLE XXXVII.

	P. P. to A.	A to B	B to C	C to D	D to E
Distance between points	200	500	1500	1500	500
Radiation supplied	125000	85000	50000	28000	12000
Volume of water passing point in cu. ft. per sec	4.74	3.27	1.75	1.	.44
Velocity f. p. s	6	6	5	5	5
Area of pipe sq. ft	.79	.545	.35	.20	.087
Diam. of pipe in ft	1	.83	.66	.5	.33
h_f by (73) for flow main	2.2	6.7	17.4	23.3	11.7
h_f (taking ⅔ value)	1.47	4.47	11.6	15.5	7.8
h_f (⅔ val. flow and return)	2.94	8.94	23.2	31.0	15.6

From the last line of the table obtain the total friction head for both mains, not including ells, tees, valves, etc., to be 81.6 feet. This is equivalent to 34.3 pounds per square inch. Allowing 20 per cent. of all the line losses to cover the minor losses we have approximately 40 pounds differential pressure, which is a reasonable value.

Another approximate method of analyzing this problem is to assume the amount of water passing any run of main to be the total requirement beyond the run plus one-half of that amount taken off through the tappings along the run and estimate the friction head from this figure. This plan will call for the full value of Equation 95 and not the two-thirds value as before. As an illustration, allowing one gallon of water per square foot of radiation per hour, approximately 125000 gallons pass from *P. P.* to *A*. Some of this is taken off in tappings and the rest of 40000 is taken off through the branch main. 85000 pass *A* and 35000 are taken off through the tappings to *B*. 50000 pass *B* and 22000 are taken off through tappings to *C*. 28000 pass *C* and 16000 are taken off to *D*. 12000 pass *D* and all are taken through tappings to *E*, the end of the line.

The amount of water chargeable to each run will be: *P. P.* to *A*, 125000, *A* to *B*, 50000 + 35000 ÷ 2 = 67500, *B* to *C*, 28000 + 22000 ÷ 2 = 39000, *C* to *D*, 12000 + 16000 ÷ 2 = 20000, and from *D* to *E*, 12000 ÷ 2 = 6000 gallons per hour. Reduced to cu. ft. per sec. this is *P. P.* to *A*, 4.7, *A* to *B*, 2.5, *B* to *C*, 1.44, *C* to *D*, .74, and *D* to *E*, .44.

For purpose of comparing with preceding method, volumes, velocities, pipe sizes (the same as in Table XXXVII) and friction heads are shown in Table XXXVIII.

TABLE XXXVIII.

	P P to A	A to B	B to C	C to D	D to E
Volume passing through section, cu. ft. per sec.	4.7	2.5	1.44	.74	.44
Average velocity in f. p. s.	6	4.6	4.1	3.7	5.
Area of pipe in sq. ft.	.79	.545	.35	.20	.087
Diam. of pipe in ft.	1	.83	.66	.5	.33
h_f by (95) flow and return	4.4	2.34	23.6	25.6	23.4

Total friction head = 79. 34 ft. not including ells, tees, valves, etc.

170. Velocity of the Water in the Mains and the Diameter of the Mains:—The district is first chosen and the layout of the conduit system is made. This is done independently of the sizes of the pipes. When this layout is finally completed, the pipe sizes are roughly calculated for all the important points in the system and are tabulated in connection with the friction losses for these parts, as in Art. 169. When this is done, Equation 96, which is recommended to be used in connection with Equation 95, may be applied and the theoretical diameters found. (The approximate diameters and the friction heads need not be calculated in Equation 95 for use in Equation 96, providing some estimate may be made for the value of h_f, for the various lengths of pipe. If desired, h_f may be assumed without any reference to the diameter, but this is a rather tedious process. For discussion of this point see Church's Hydraulic Motors, Arts. 121-124 b.)

$$d = .629 \left[\tfrac{2}{3} \times \frac{\phi l Q^2}{h_f} \right]^{\frac{1}{5}} \tag{96}$$

where d, h_f, ϕ and l are the same as in Equation 95, and Q = cubic feet of water passing through the pipe per second. This equation differs from those given in the references stated, in that the term ⅔ is inserted as a mean value between the two extreme conditions, as stated in Art. 169.

APPLICATION.—Let it be desired to find the diameter for the single main between the power plant and A, Art. 169, with $h_f = 1.47$

$$d = .629 \left[\frac{2 \times .005 \times 200 \times (4.74)^2}{3 \times 1.47} \right]^{1/5} = 1 \text{ ft.} = 12 \text{ in.}$$

Applying to the entire line with h_f as given in next to last line of Table XXXVII, gives power plant to A, $d = 12$ inches; A to B, $d = 10$ inches; B to C, $d = 8$ inches; C to D, $d = 6$ inches; and D to E, $d = 4$ inches.

In some cases, when close estimating is not required, it is satisfactory to assume a velocity of the water and find the diameter without considering the friction loss. In many cases, however, this would soon prove a positive loss to the company. With a low velocity, the first cost would be large and the operating cost would be low. On the other hand, if the velocity were high, the first cost would be small and the operating cost and depreciation would be large. As an illustration of how the friction head increases in a pipe of this kind with increased velocity, refer to the run of mains between B and C. Assuming a velocity of 10 feet per second, which in this case would be very high, the friction head, h_f, for the single main, becomes 62 and the theoretical diameter is 5.5, say 6 inches. The friction head, as will be seen, is 5.4 times the corresponding value when the velocity was 5 feet per second. Since the pump must work continually against this head, it would incur a financial loss that would soon exceed the extra cost of installing larger pipes. It is found in plants that are in first class operation that the velocities range from 5 to 7 feet per second.

The calculations in Arts. 169 and 170 are very much simplified by the use of the chart shown in the Appendix. In planning a system of this kind, find the friction head on the pumps and the diameters of the pipes for various velocities, say 4, 6, 8 and 10 feet per second. Estimate the probable first cost and the depreciation of the conduit system for each velocity, and balance these figures with the operating cost for a period of, say five years, to see which is the most economical velocity to use in figuring the system.

171. Service Connections are usually installed from 30 to 36 inches below the surface of the ground, and are insulated in some form of box conduit which compares favorably with that of the main conduit. Service branches are

1¼-, 1½- and 2-inch wrought iron pipe. These are usually carried to the building from the conduit at the expense of the consumer. Such branch conduits are not drained by tile drains.

172. Total Steam Available and B. t. u. Liberated per Hour for Heating the Circulating Water:—The amount of steam available for heating the circulating water is that given off by the generating units, plus that from the circulating pumps, plus that from the city water supply pumps if there be any, plus that from the auxiliary steam units in the plant, i. e., small pumps, engines, etc. In the typical application this amounts to $23100 + 12720 + 8680 = 44500$ pounds per hour.

This steam, of course, is not equal to good dry steam in heating value because of the work it has done in the engine and pump cylinders, but a good estimate of its value may be approximated. In addition to the terms used in Equation 93, let q' = heat in the returning condensation per pound; then the heat available for heating purposes per pound of exhaust steam is

$$\text{B. t. u.} = x\,r + q - q' \qquad (97)$$

It is probably safe to consider the quality of the steam as 85 per cent. of that of good dry steam at the same pressure. Since the pressure of the exhaust from a non-condensing engine, as it enters the heater, is near that of the atmosphere, and since the returning condensation is at a temperature of about 180°, the total amount of heat given off from a pound of exhaust steam to the circulating water is $.85 \times 970.4 + 180 - (180 - 32) = 856.84$, say 850. If W_s = pounds of exhaust steam available, the total number of B. t. u. given off from the exhaust steam per hour is

$$\text{Total B. t. u.} = 850\, W_s \qquad (98)$$

Applying this to the typical power plant gives $850 \times 44500 = 37825000$ B. t. u. per hour. This amount is probably a maximum under the conditions of lighting units as stated, and would be true for only 5 hours out of 24. At other times the exhaust steam drops off from the lighting units and this deficiency must be made good by heating the circulating water directly from the coal, by passing the water through heating boilers or by passing it through economizers where it is heated by the waste heat from the stack gases.

173. Amount of Hot Water Radiation in the District that can be Supplied by One Pound of Exhaust Steam on a Zero Day:—In Art. 166, each pound of water takes on 25 B. t. u. in passing through the reheaters at the power plant, and gives off at least 20 B. t. u. in passing through the radiator. The number of pounds of water heated per pound of steam per hour is, W_w = (Total B. t. u. available per pound of exhaust steam per hour) $\div$ 25, and the total radiation supplied is

$$R_w = \frac{\text{Total B. t. u. available per lb. of exhaust steam per hr.}}{8.33 \times 25} \qquad (99)$$

which for average practice reduces to

$$R_w = \frac{850}{208} = 4 \text{ square feet approx.} \qquad (100)$$

Applying Equation 99 for the five hour period when the exhaust steam is maximum gives $R_w = 37825000 \div 208 =$ 181851 square feet. It is not safe to figure on the peak load conditions. It is better to assume that for half the time, 35000 pounds of steam are available and will heat $35000 \times 4 = 140000$ square feet of radiation.

174. The Amount of Circulating Water Passed through the Heater Necessary to Condense One Pound of Exhaust Steam is

$$W_w = \frac{\text{Total B. t. u. available per lb. of exhaust steam per hr.}}{25} \qquad (101)$$

With the value given above for the exhaust steam this becomes, for 100 and 85 per cent. respectively,

$$W_w = \frac{1000}{25} = 40 \text{ pounds} \qquad (102)$$

$$W_w = \frac{850}{25} = 34 \text{ pounds} \qquad (103)$$

175. Amount of Hot Water Radiation in the District that can be Heated by One Horse-Power of Exhaust Steam from a Non-Condensing Engine on a Zero Day:—

$$R_w = 4 \times (\text{pounds of steam per } H.\ P. \text{ hour}) \qquad (104)$$

This reduces for the various types of engines, as follows:

Simple high speed	$4 \times 34 = 136$	square feet.			
" medium "	$4 \times 30 = 120$	"	"		
" Corliss	$4 \times 26 = 104$	"	"		
Compound high "	$4 \times 26 = 104$	"	"		
" medium "	$4 \times 25 = 100$	"	"		
" Corliss	$4 \times 22 = 88$	"	"		

176. Amount of Radiation that can be Supplied by Exhaust Steam in Equations 99 and 100 at any other Temperature of the Water, t_w, than that Stated, with the Room Temperature, t', Remaining the Same:—The amount of heat passing through one square foot of the radiator to the room is in proportion to $t_w - t'$. In Equations 99 and 100, $t_w - t' = 100$. Now if t_w be increased x degrees, so that $t_w - t' = (100 + x)$ then each square foot of radiation in the building will give off $\frac{100 + x}{100}$ times more heat than before and each pound of exhaust steam will supply only

$$R_w = \frac{4 \times 100}{100 + x} \text{ square feet} \qquad (105)$$

This for an increase of 30 degrees, which is probably a maximum, is

$$R_w = \frac{4}{1.3} = 3 \text{ square feet} \qquad (106)$$

Compared with Equation 100, Equation 105 shows, with a high temperature of the water entering the radiator, that less radiation is necessary to heat any one room and that each square foot of surface becomes more nearly the value of an equal amount of steam heating surface. Calculations for radiation, however, are seldom made from high temperatures of the water, and this article should be considered an exceptional case.

177. Exhaust Steam Condenser (Reheater), for Reheating the Circulating Water:—In the layout of any plant the reheaters should be located close to the circulating

pumps on the high pressure side. They are usually of the surface condenser type (Fig. 167) and may or may not be installed in duplicate. Of the two types shown in the figure, the water tube type is probably the more common. The same principles hold for each in design. In ordinary heaters for feed water service, wrought iron tubes of 1½- to 2-inches

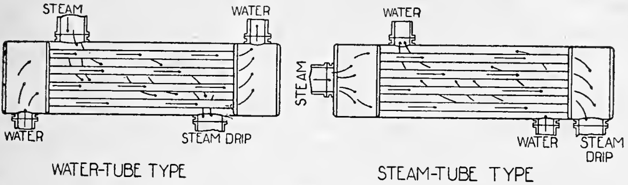


Fig. 167.

diameter are generally used, but for condenser work and where a rapid heat transmission is desired, brass or copper tubes are used, having diameters of ⅝- to 1-inch. In heating the circulating water for district service, the reheater is doing very much the same work as if used on the condensing system for engines or turbines. The chief difference is in the pressures carried on the steam side, the reheater condensing the steam near atmospheric pressure and the condenser carrying about .9 of a perfect vacuum. In either case it should be piped on the water side for water inlet and outlet, while the steam side should be connected to the exhaust line from the engines and pumps, and should have proper drip connection to draw the water of condensation off to a condenser pump. This condenser pump usually delivers the water of condensation to a storage tank for use as boiler feed, or for use in making up the supply in the heating system.

In determining the details of the condenser the following important points should be investigated: the amount of heating surface in the tubes, the size of the water inlet and outlet, the size of the pipe for the steam connection, the size of the pipe for the water of condensation and the length and cross section of the heater.

178. Amount of Heating Surface in the Reheater Tubes:—The general equation for calculating the heating surface in

the tubes of a reheater (assuming all heating surface on tubes only), is

$$R_t = \frac{\text{Total B. t. u. given up by the exhaust steam per hr.}}{K\ (\text{Temp. diff. between inside and outside of tubes})} \qquad (107)$$

The maximum heat given off from one pound of exhaust steam in condensing at atmospheric pressure is 1000 B. t. u., the average temperature difference is approximately 47 degrees, and K may be taken 427 B. t. u. per degree difference per hour. In determining K, it is not an easy matter to obtain a value that will be true for average practice. Carpenter's H. & V. B. Art. 47 quotes the above figure for tests upon clean tubes, and volumes of water less than 1000 pounds per square foot of heating surface per hour. It is found, however, that the *average* heater or condenser tube with its lime and mud deposit will reduce the efficiency as low as 40 to 50 per cent. of the maximum transmission. Assume this value to be 45 per cent.; then if W_s is the number of pounds available exhaust steam, Equation 107 becomes

$$R_t = \frac{1000\ W_s}{K\ (t_s - t_w)} = \frac{1000\ W_s}{427 \times .45 \times 47} = \frac{1000\ W_s}{9031} = \frac{W_s}{9.1} \qquad (108)$$

In "Steam Engine Design," by Whitham, page 283, the following equation is given for surface condensers used on shipboard:

$$S = \frac{W\ L}{cK\ (T_1 - t)}$$

where S = tube surface, W = total pounds of exhaust steam to be condensed per hour; L = latent heat of saturated steam at a temperature T_1, K = theoretical transmission of B. t. u. per hour through one square foot of surface per degree difference of temperature = 556.8 for brass, c = efficiency of the condensing surface = .323 (quoted from Isherwood), T_1 = temperature of saturated steam in the condensers, and t = average temperature of the circulating water.

With $L = 970.4$, $c = .323$, $K = 556.8$ and $T_1 - t = 47$, we may state the equation in terms of our text as

$$R_t = \frac{970.4\ W_s}{.323 \times 556.8 \times 47} = \frac{970.4\ W_s}{8446} = \frac{W_s}{8.7} \qquad (109)$$

In Sutcliffe "Steam Power and Mill Work," page 512, the author states that condenser tubes in good condition and set

in the ordinary way have a condensing power equivalent to 13000 B. t. u. per square foot per hour, when the condensing water is supplied at 60 degrees and rises to 95 degrees at discharge, although the author gives his opinion that a transmission of 10000 B. t. u. per square foot per hour is all that should be expected. This checks closely with Equation 108, which gives the rate of transmission 9031 B. t. u. per square foot per hour.

The following empirical equation for the amount of heating surface in a heater is sometimes used:

$$R_t = .0944\ W_s \tag{110}$$

where the terms are the same as before.

APPLICATION.—Let the total amount of exhaust steam available for heating the circulating water be 35000 pounds per hour, the pressure of the steam in the condenser be atmospheric and the water of condensation be returned at 180°; also, let the circulating water enter at 155° and be heated to 180°. These values are good average conditions. The assumption that the pressure within the condenser is atmospheric might not be fulfilled in every case, but can be approached very closely. From these assumptions find the square feet of surface in the tubes.

$$\text{Equation 108, } R_t = \frac{35000}{9.1} = 3846 \text{ sq. ft.}$$

$$\text{Equation 109, } R_t = \frac{35000}{8.7} = 4023 \text{ sq. ft.}$$

$$\text{Equation 110, } R_t = 35000 \times .0944 = 3304 \text{ sq. ft.}$$

$$\text{Sutcliffe} \qquad R_t = \frac{1000 \times 35000}{10000} = 3500 \text{ sq. ft.}$$

If 3846 square feet be the accepted value it will call for three heaters having 1282 square feet of tube surface each.

179. Amount of Reheater Tube Surface per Engine Horse-Power:—Let w_s be the pounds of steam used per *I. H. P.* of the engine; then from Equation 108

$$R_t \text{ (per } I.\ H.\ P.) = \frac{w_s}{9.1} \tag{111}$$

This reduces for the various types of engines as follows:

Simple high speed	$34 \div 9.1$	$= 3.74$	square feet
" medium "	$30 \div 9.1$	$= 3.30$	" "
" Corliss	$26 \div 9.1$	$= 2.86$	" "
Compound high "	$26 \div 9.1$	$= 2.86$	" "
" medium "	$25 \div 9.1$	$= 2.75$	" "
" Corliss	$22 \div 9.1$	$= 2.42$	" "

180. Amount of Hot Water Radiation in the District that can be Supplied by One Square Foot of Reheater Tube Surface:—If the transmission through one square foot of tube surface be $K\ (t_s - t_w) = 9031$ B. t. u. per hour and the amount of heat needed per square foot of radiation per hour $= 8.33 \times 25 = 208$, as given in Eqaution 99, then

$$R_w \text{ (per sq. ft. of tube surface)} = \frac{9031}{208} = 43.4 \text{ sq. ft.} \quad (112)$$

181. Some Important Reheater Details:—*Inlet and outlet pipes.*—Having three heaters in the plant, it seems reasonable that each heater should be prepared for at least one-third of the water credited to the exhaust steam. From Art. 173 this is $140000 \div 3 = 46667$ gallons $= 10800000$ cubic inches per hour. The velocity of the water entering and leaving the heater may vary a great deal, but good values for calculations may be taken between 5 and 7 feet per second. Assuming the first value given, we have the area of the pipe $= 10800000 \div (5 \times 12 \times 3600) = 50$ square inches, and the diameter 8 inches.

The size of the reheater shell.—Concerning the velocity of the water in the reheater itself, there may be differences of opinion; 100 feet per minute will be a good value to use unless this value makes the length of the tube too great for its diameter. If this is the case the tube will bend from expansion and from its own weight. At this velocity the free cross sectional area of the tubes, assuming the water to pass through the tubes as in Fig. 167, will be 150 square inches. If the tubes be taken ¾ inch outside diameter, with a thickness of 17 B. W. G., and arranged as usual in such work, it will require about 475 tubes and a shell diameter of approximately 30 inches. If the inner surface of the tube be taken as a measurement of the heating surface and

the total surface be 1282 square feet, the length of the reheater tubes will be approximately 16 feet.

The ratio of the length of the tube to the diameter is, in this case, 256, about twice as much as the maximum ratio used by some manufacturers. It will be better, therefore, to increase the number of tubes and decrease the length. With a velocity of the water of 50 feet per minute, the values will be approximately as follows: free cross sectional area of the tubes, 300 square inches; number of tubes, 950; diameter of shell, 40 inches; length of tubes, 8 feet. These values check fairly well and could be used.

The size of exhaust steam connection.—To calculate this, use the equation

$$A = \frac{144\ Q_s}{V} \tag{113}$$

where Q_s = volume of steam in cubic feet per minute, V = velocity in feet per minute, and A = area of pipe in square inches. When applied to the reheater using 35000 pounds of steam per hour, at 26 cubic feet per pound and at a velocity through the exhaust pipe of 6000 feet per minute, it gives

$$A = \frac{144 \times 35000 \times 26}{60 \times 6000} = 360 \text{ sq. in.} = 22 \text{ in. dia.}$$

Try also, from Carpenter's H. & V. B., page 284

$$d = \sqrt[5]{\frac{H\ P^2}{1.23}} \tag{114}$$

Allowing 30 pounds of steam per *H. P.* hour for non-condensing engines we have 35000 ÷ 30 = 1166 horse-power; then applying the above we obtain d = 16 inches. Comparing the two Equations, 113 and 114, the first will probably admit of a more general application. The velocity 6000 for exhaust steam may be increased to 8000 for very large pipes and should be reduced to 4000 for small pipes. In the above applications a 20 inch pipe will suffice.

The return pipe for condensation.—The diameter of the pipe leading to the condenser pump will naturally be taken from the catalog size of the pump installed. This pump would be selected from capacities as guaranteed by the respective manufacturers and should easily be capable of handling the amount of water that is condensed per hour.

The value of a high pressure steam connection.—If desired, the reheater may also be provided with a high pressure

steam connection, to be used when the exhaust steam is not sufficient. This steam is then used through a pressure-reducing valve which admits the steam at pressures varying from atmospheric to 5 pounds gage. There is some question concerning the advisability of doing this. Some prefer to install a high pressure steam heater, as in Art. 182, to be used independently of the exhaust steam heaters. This removes all possibility of having excessive back pressure on the engine piston, as is sometimes the case where high pressure steam is admitted with the exhaust steam.

It has been the experience of some who have operated such plants that where more heat is needed than can be supplied by the exhaust steam, it is better to resort to heating boilers and economizers ,than to use high pressure steam for heating.

182. High Pressure Steam Heater:—When this heater is used it is located above the boiler so that all the condensa-

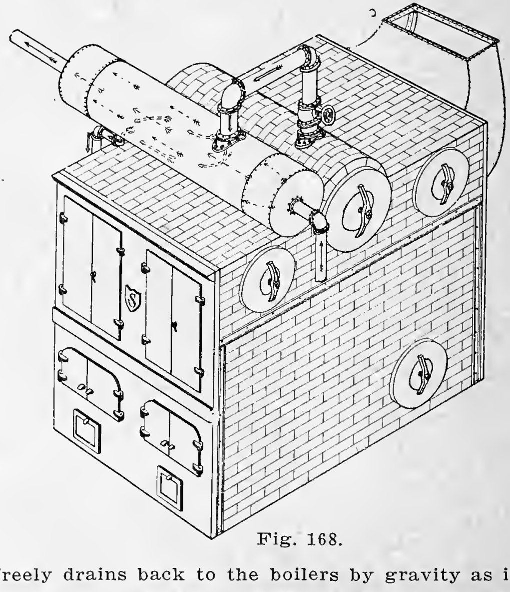


Fig. 168.

tion freely drains back to the boilers by gravity as in Fig. 168. In calculating the tube surface, use Equation 107 with

the full value of the steam and the steam temperatures changed to suit the increased pressure. Such a heater as this gives good results.

183. Circulating Pumps:—Two types of pumps are in general use: centrifugal and reciprocating. Each type is somewhat limited in its operation. The centrifugal pump has difficulty in operating against high heads and the reciprocating pump is very noisy when running at a high piston speed. Since each type is in successful operation in many plants, no comparisons will be made between them further than to say that the former, being operated by a steam engine, may be run more economically than the latter because of the possibilities of using the steam expansively. It will be noted, however, that this same steam is to be used in the exhaust steam heaters for warming the circulating water and hence there would be little, if any, direct loss from this source in the use of the reciprocating pump.

Having given the maximum amount of water to be circulated per hour, consult trade catalogs and select the number of pumps and the size of each pump to be installed. The sizes of the pumps can easily be determined when the number of them has been decided upon. This latter point is one upon which a difference of opinion will probably be found. No exact rule can be applied. In a plant of, say not more than 150000 square feet of radiation (150000 gallons of water per hour, or 3 million gallons for twenty-four hours), some designers would put in three pumps, each having 1.5 million gallons capacity; in which case one pump could be cut out for repairs and the other two would be able to care for the service temporarily. Other designers would use four pumps at about one million gallons each. The fewer the pumps installed, in any case, the greater should be the capacity of each. The following values will be found fairly satisfactory:

1 Pump. Cap.	= (1 to 1.25)	times	max.	requirem't	of	system
2 Pumps. "	(each) = .75	"	"	"	"	"
3 Pumps. "	" = .5	"	"	"	"	"
4 Pumps. "	" = .3	"	"	"	"	"

Having given the capacity of each pump in gallons of water per minute, the size, the horse-power and the steam consumption of each pump can be calculated. In obtaining the size of the pump it will be necessary to know the *speed*,

V, of the piston in feet per minute, the *strokes*, N, per minute and the per cent. of *slip*, s (100 *per cent.* — S, where S = hydraulic efficiency). The speed varies between 100, for small pumps, and 75, for large pumps. The strokes vary between 200, for small pumps, and 40, for large pumps, and the slip varies between 5 and 40 per cent., depending upon the fit of the piston and the valves. In pumps that have been in service for some time the slip will probably average 20 per cent.

The cross sectional area of the water cylinder in square inches is

$$W.\ C.\ A. = \frac{\text{cubic inches pumped per minute}}{S \times V \times 12} \qquad (115)$$

from which we may obtain the diameter of the water cylinder.

The steam cylinder area is usually figured as a certain ratio to that of the water cylinder area, as

$$S.\ C.\ A. = (1.5 \text{ to } 2.5) \times W.\ C.\ A \qquad (116)$$

from which we may obtain the diameter of the steam cylinder.

The length of the stroke, L, in inches, may be obtained from the speed and the number of strokes such that

$$L = \frac{12\ V}{N} \qquad (117)$$

All direct acting steam pumps are designated by diameter of steam cylinder × diameter of water cylinder × length of stroke, as

$$14'' \times 12'' \times 18''$$

Duplex pumps have twice the capacity of single pumps having the same sized cylinders.

To find the indicated horse-power, *I. H. P.*, of the pumps, reduce the pressure head, p, in pounds per square inch, to pressure head in feet, h; multiply this by the pounds of water, W, pumped per minute and divide the product by 33000 times the mechanical efficiency, E.

$$I.\ H.\ P. = \frac{W\ h}{33000\ E} \qquad (118)$$

To reduce from pressure head in pounds to pressure head in feet, divide the pressure head in pounds by weight

of a column of water one square inch in area and one foot high. The general equation for this is

$$h = \frac{144\ p}{w}$$

where w = the weight of a cubic foot of water at the given temperature and p = differential pressure in pounds per square inch.

In pump service of this kind the pressure head, p, against which the pump is acting, is not the result of the static head of water in the system but is due to the inertia of the water and to the resistance to the flow of water through the piping system and the heaters. This frictional resistance may be calculated as shown in Art. 169. Read this part of the work over carefully.

For an illustration of combined pressure head, p, and friction head, h_f, see Art. 186 on boiler feed pumps. Having found the *I. H. P.* of any pump, multiply it by the steam consumption per *I. H. P.* hour and the result will be the steam consumption of the pump. This exhaust steam will be considered a part of the general supply when figuring the size of the exhaust steam heaters in the system.

The mechanical efficiency, E, of piston pumps depends upon the condition of the valves and upon the speed, and varies from 90 per cent. in new pumps, to 50 per cent. in pumps that are badly worn. A fair average would be 70 per cent.

The steam consumption for reciprocating, simple and duplex non-condensing pumps would approximate 100 to 200 pounds of steam per *I. H. P.* hour—the greater values referring to the slower speeds.

184. Centrifugal Pumps:—Centrifugal pumps are of two classifications, the Volute and the Turbine. The principles upon which each operate are very similar. The rotating impeller, or rotor, with curved blades draws the water in at the center of the pump and delivers it from the circumference. The rotor is enclosed by a cast iron casement which is shaped more or less to fit the curvature of the edges of the blades on the rotor. Centrifugal pumps are used where large volumes of water are required at low heads. They are used in city water supply systems, in central station heating systems, in condenser service, in irri-

gation work and in many other places where the pressure head operated against is not excessive. The efficiency of the average centrifugal pump is from 65 to 80 per cent., 75 per cent. being not uncommon. In places where such pumps are used the head is usually below 75 feet, although some types, when direct connected to high speed motors, are capable of operating against heads of several hundred feet.

Some of the advantages of centrifugal pumps over horizontal reciprocating pumps are: low first cost, simplicity, few moving parts, compactness, uniform flow and pressure of water, freedom from shock, possibilities of direct connection to high speed motors and the ability to handle dirty water without injuring the pump.

One of the advantages of piston pumps over centrifugal pumps is the fact that they are more positive in their operation and work against higher heads.

Centrifugal pumps, when connected to engine and turbine drives, benefit by the expansion of the steam and are much more economical than the direct acting piston pump, which takes steam at full pressure for nearly the entire stroke. The amount of steam used in the pumps in central station work, however, is not a serious factor, since all of the heat in the steam that is not used in propelling the water through the mains is used in the heaters to increase the temperature of the water.

The sphere of usefulness of the centrifugal pump in central station heating is increasing. The direct acting piston pump, when operating at fairly high speeds, causes hammering and pounding in the transmission lines, and these noises are sometimes conveyed to the residences and become annoying to the occupants. This feature is not so noticeable in the operation of the centrifugal pump.

Application.—In Art. 167 assume the capacity of the plant, 10 business blocks and 21 residence blocks, to require 184500 gallons of water per hour; the same to be pumped against a pressure head (Art. 169) of 50—5 pounds, by horizontal, direct acting piston pumps. Assume also the steam consumption of the pumps to be 100 pounds per *I. H. P.* hour and the average temperature of the water at the pumps to be $(180 + 155) \div 2 = 167.5$ degrees. Apply Equation 118, where h = calculated total friction head for the longest line

in the system (this is designated by h_f in Art. 169), or where p = total difference between the incoming and the outgoing pressures. With the weight of a cubic foot of water at 167.5 degrees = 60.87 pounds and with $p = 45$, we have $h = 106.5$ feet, and the indicated horse-power of the pumps, assuming 65 per cent. mechanical efficiency, is

$$I.\ H.\ P. = \frac{184500 \times 8.33 \times 106.5}{33000 \times .65 \times 60} = 127.2$$

From this the steam consumption will probably be 12720 pounds per hour.

If centrifugal pumps were selected the horse-power would be calculated from the same equation, but the steam consumption of the engine driving it would probably be 30 to 40 pounds of steam per horse-power.

185. City Water Supply Pumps:—Horizontal, direct acting duplex pumps for use on city water supply service are the same as those used to circulate the water in heating systems; hence, the foregoing descriptions apply here. The *I. H. P.* of the city water supply pumps would be calculated by use of Equation 118. If the pumps lift the water from the wells, as would probably be the case, the suction pressure would be negative and would be added to the force pressure.

Application.—Assume the pressure in the fresh water mains 60 pounds and the suction pressure 10 pounds; therefore, $p = 60 - (-10) = 70$ pounds, and with the water at 65 degrees, $h = 144 \times 70 \div 62.5 = 161$ feet. These pumps are each rated at 1.5 million gallons in 24 hours, and deliver $62500 \times 8.33 = 520833$ pounds of water per hour, when running at full capacity. Assuming each pump to deliver 75 per cent. of the full requirement of the system, the total amount of water pumped per hour for the city water supply would approximate $520833 \div .75 = 694444$ pounds, and the total average horse-power used in pumping the water would be

$$I.\ H.\ P. = \frac{694444 \times 161}{60 \times 33000 \times .65} = 86.8$$

With 100 pounds of steam per horse-power hour, this would amount to 8680 pounds of exhaust steam available per hour for use in heating the circulating water.

186. Boiler Feed Pumps:—Horizontal pumps for high pressure boiler feeding are selected in a similar way. Such units are called auxiliary steam units and, because the steam

required is small, they are sometimes piped to a feed water heater for heating the boiler feed. The velocity of the water through the suction pipe may be taken 200 feet per minute and in the delivery pipe 300 feet per minute. The piston speed, the strokes per minute and the slip would be very much the same as stated under circulating pumps. Such pumps should have a pumping capacity about twice as great as the actual boiler requirements, and in small plants where only one pump is needed, the installation should be in duplicate. The sizes of the cylinders and the efficiencies are about as stated for the larger circulating pumps.

In determining the horse-power of a boiler feed pump, four resistances must be overcome; i. e., pressure head, p, or boiler pressure; suction heat, h_s; delivery head, h_d, and the friction head, h_f. The first three values are usually given. The friction head includes the resistances in all piping, ells and valves from the supply to the boiler. The friction in the piping may be taken from Table 42, Appendix, or it may be worked out by Equation 95. The friction in the ells and valves is more difficult to determine and is usually stated in equivalent length of straight pipe of the same diameter. A rough rule used by some in such cases is as follows: "to the length of the given pipe, add 60 times the nominal diameter of the pipe for each ell, and 90 times the diameter for each globe valve," then find the friction head as stated above. A straight flow gate or water valve could safely be taken as an ell. For simplicity of calculation, all of the above resistances may be reduced to an equivalent head, such that

$$h_e = \frac{144\,p}{w} + h_d + h_s + h_f \tag{119}$$

where w = weight of one cubic foot of water at the suction temperature, w may be obtained from Table 9, Appendix, and h_f may be taken from Table 42. The horse-power by Equation 118 then becomes, if W = pounds of water pumped per minute,

$$I.\,H.\,P. = \frac{W \times h_e}{33000\,E} \tag{120}$$

Application.—Let p = 125 pounds gage, w = 62.5, h_d = 8 feet, h_s = 20 feet, horizontal run of pipe from supply to

pump = 20 feet, horizontal run of pipe from pump to boiler = 30 feet; also, let the pump supply 89000 pounds of water per hour to the boiler. This is twice the capacity of the boiler plant. With this amount of water at the usual velocity it will give a suction pipe of 4.5 inches diameter, and a flow pipe of 4 inches diameter. Let there be two ells and one gate valve on the suction pipe, and three ells, one globe valve and one check valve on the delivery pipe. We then have an equivalent of 107 feet of suction pipe, and 158 feet of delivery pipe. Referring to Table 42, h_f is approximately 7 feet, and the total head is

$$h_e = \frac{144 \times 125}{62.5} + 8 + 20 + 7 = 323 \text{ feet.}$$

In the use of most boiler feed pumps it is considered unnecessary to determine h_f so carefully. A very satisfactory way is to obtain the total head pumped against, exclusive of the friction head, and add to it 5 to 15 per cent., depending upon the complications in the circuit. Substituting the above in Equation 120, we obtain

$$I.\ H.\ P. = \frac{89000 \times 323}{60 \times 33000 \times .65} = 22.3$$

Work out the value of h_f by Equation 95 and see how nearly it checks with the above.

187. Boilers:—A number of boilers will necessarily be installed in a plant of this kind, and a good arrangement is to have them so piped with water and steam headers that any number of the boilers may be used for steaming purposes and the rest as water heaters. They should also be so arranged that any of the boilers may be thrown out of service for cleaning or repairs and still carry on the work of the plant. By doing this the boiler plant becomes very flexible and each boiler is an independent unit. Any good water tube boiler would serve the purpose both as a steaming and as a heating boiler. Where the boilers are used as heaters, the water should enter at the bottom and come out at the top. Where the water enters at the top and comes out at the bottom, the excessive heating of the front row of tubes retards the circulation of the water by this heat, and produces a rapid circulation through the rear tubes where the heat is the least. This rapid circulation in the rear tubes

is not a detriment, but it is less needed there than in the front ones. It would be decidedly better if the rapid circulation were in the front row, causing the heat from the fire to be carried off more readily, and by this means giving less danger of burning the tubes. In the latter case the forced circulation from the pumps will be aided by the natural circulation from the heat of the fire, and the life of all the tubes becomes more uniform. Fig. 169 shows a typical header arrangement.

Boilers are usually classified as fire tube and water tube. *Fire tube boilers* are of the multitubular type, having the flue gases passing through the tubes and water surrounding them. *Water tube boilers* have the water passing through the tubes and the flue gases surrounding them. The *heating surface* of a boiler is composed of those boiler plates having the heated flue gases on one side and the water on the other. A *boiler horse-power* may be taken as follows:

Centennial Rating.

One *B. H. P.* = 30 pounds of water evaporated from feed water at 100° F. to steam at 70 pounds gage pressure.

A. S. M. E.

One *B. H. P.* = 34.5 pounds of water evaporated from and at 212° F.

In laying out a boiler plant some good approximations for the essential details are:

One *B. H. P.* = 11.5 square feet of heating surface. (multitubular type).

One *B. H. P.* = 10 square feet of heating surface (water tube type).

One *B. H. P.* = .33 square foot of grate surface (small plant, say one boiler).

One *B. H. P.* = .25 square foot of grate surface (medium sized plant, say 500 H. P.).

One *B. H. P.* = .20 square foot of grate surface (large plants).

Pounds of water evaporated per square foot of heating surface per hour = 3 (approx. value).

188. Square Feet of Hot Water Radiation that can be Supplied on a Zero Day by One Boiler Horse-Power when the Boiler is Used as a Heater:—Assuming that the coal used in the plant has a heating value of 13000 B. t. u. per pound,

and that the efficiency of the boiler is 60 per cent., each pound of coal will transmit to the water 7800 B. t. u. Since each pound of water takes up 25 B. t. u. on its passage through the heating boiler, one pound of coal will heat 312 pounds, or 37.5 gallons of water. This is equivalent to supplying heat, under extreme conditions of heat loss, to 37.5 square feet of radiation for one hour. One boiler horse-power, according to Art. 187, is equivalent to the expenditure of $970.4 \times 34.5 = 33478$ B. t. u. Now since each pound of coal transfers to the water 7800 B. t. u., one boiler horse-power will require $33478 \div 7800 = 4.29$ pounds of coal. If, then, the burning of one pound of coal will supply 37.5 square feet of hot water radiation for one hour, one boiler horse-power will supply $4.29 \times 37.5 = 160$ square feet for one hour, and a 100 H. P. boiler will supply 16000 square feet of water radiation in the district for the same time. These figures have reference to boilers under good working conditions and probably give average results.

189. Square Feet of Hot Water Radiation in the District that can be Supplied on a Zero Day by an Economizer Located in the Stack Gases between the Boilers and the Chimney:—In order to make this estimate it is necessary first to know the horse-power of the boilers, the amount of coal burned per hour, the pounds of gases passing through the furnace per hour and the heat given off from these gases to the circulating water through the tubes.

Application.—Let C = pounds of coal burned per hour = boiler horse-power $\times$ pounds of coal per boiler horse-power hour, W_a = pounds of air passed through the furnace per pound of fuel burned, s = specific heat of the gases, t_b = temperature of gases leaving boiler, t_s = temperature of gases leaving economizer, t_w = temperature of water entering economizer and t_f = temperature of water leaving the economizer. Then, if 8.33 pounds of water will supply one square foot of radiation for one hour we have

$$R_w = \frac{s \times (C \times W_a + C) \times (t_b - t_s)}{8.33 \times (t_f - t_w)} \qquad (121)$$

In the illustrative plant, 44500 pounds of steam per hour are generated in the steam boiler plant at a pressure of 125 pounds gage. To find the boiler horse-power let the total heat of the steam, above 32° at 125 pounds gage, be 1192 B. t. u., and let the temperature of the incoming feed water

to the boilers be 60 degrees. (In most cases the feed water will be at a higher temperature, but since it will occasionally be as low as 60 degrees, this value should be used.) The heat put into a pound of steam under these conditions is 1192 — (60 — 32) = 1164 B. t. u., and in 44500 pounds it will be 51798000 B. t. u. Since one horse-power of boiler service is equivalent to 33455 B. t. u., we will need 51798000 ÷ 33478 = 1548 boiler horse-power. This horse-power will take care of all the engines and pumps in the plant. If the coal used contains 13000 B. t. u. per pound and the boilers have 60 per cent. efficiency, 7800 B. t. u. will be given to the water per pound of fuel burned, and the amount of coal burned per hour will be 51798000 ÷ 7800 = 6640 pounds. This gives 6640 ÷ 1548 = 4.3 pounds of fuel per boiler horse-power hour, and 6.7 pounds of water evaporated per pound of fuel. If the flue gases have 12 per cent. CO_2, there are used according to experimental data, about 21 pounds of air or 22 pounds of the gases of combustion, per pound of fuel burned. This is equivalent to 6640 × 22 = 146080 pounds of flue gases total. Suppose now that these gases leave the furnace for the chimney at a temperature of 550 degrees F., that the economizer drops the temperature of the gases down to 350 degrees (a condition which is very reasonable) and that the specific heat of the gases is about .22, we have 146080 × .22 × (550 — 350) = 6427520 B. t. u. given off from the gases per hour in passing through the economizer (see numerator in Equation 121). This heat is taken up by the circulating water in passing through the economizer toward the outgoing main. Now if the water as it returns from the circulating system, enters the economizer at 155 degrees, and leaves at 180 degrees, we will have 6427520 ÷ (180 — 155) = 257100 pounds of water heated per hour. This is equivalent to supplying 257100 ÷ 8.33 = 30864 square feet of radiation per hour when the plant is running at its peak load. Taking the "pounds of steam per hour" in the above as the only variable quantity, we are fairly safe in saying that the heat in the chimney gases from one horse-power of steaming boiler service will supply, through an economizer, 30864 ÷ 1548 = 20 square feet of radiation in the district. In plants where only 7 pounds of water are allowed to each square foot of radiation per hour, this becomes 23.8 square feet of radiation instead.

190. Square Feet of Economizer Surface Required to Heat the Circulating Water in Art. 189:—Let K = the coefficient of heat transmission through clean cast iron tubes and E = the efficiency of the tube surface when in average service, also let the terms for the temperatures of the gases and the circulating water be as given in Art. 189, then

$$R_e = \frac{\text{Heat trans. per hour from gases to water}}{K \times E \times \left(\dfrac{t_b + t_s}{2} - \dfrac{t_f + t_w}{2} \right)} \qquad (122)$$

This equation assumes that the rate of heat flow through the tubes is the same at all points. As a matter of fact this rate changes slightly as the water becomes heated, but the error is not serious in an equation, where the efficiency of the surface may be anything from 100 per cent. in new tubes to as low as 30 to 40 per cent. for old ones.

Application.—Let K = 7 and E = .4, then

$$R_e = \frac{6427520}{7 \times .4 \times \left(\dfrac{550 + 350}{2} - \dfrac{180 + 155}{2} \right)} = 8125 \text{ sq. ft.}$$

With 12 square feet of surface per tube this gives 677 tubes.

191. Square Feet of Economizer Surface to Install when the Economizer is to be Used to Heat the Feed Water for the Steaming Boilers:—If 30 pounds of feed water are fed to the boiler per horse-power hour, and if K = 7, E = .4, t_b = 550, t_s = 350, t_f = 250, and t_w = 90 (about the lowest temperature at which water should enter the economizer), then the square feet of surface per horse-power is

$$R_e = \frac{30 \times (250 - 90)}{7 \times .4 \times \left(\dfrac{550 + 350}{2} - \dfrac{250 + 90}{2} \right)} = 6.1 \text{ sq. ft.}$$

192. Total Capacity of the Boiler Plant and the Number of Boilers Installed:—The following discussion on the size of the boiler plant is purely for illustrative purposes and is intended to show how such problems may be analyzed. In most cases the exhaust steam, and the economizer, if used, will fall far short of supplying the total radiation in the district, especially when the electrical output is light and the weather is cold. Suppose it be desired to install extra boilers to be used as heaters for the radiation not supplied

from these two sources. To determine the amount of extra boilers, find the amount of radiation to be supplied by the exhaust steam and the economizer and subtract this from the total radiation. The difference must be supplied by boilers used as heaters. It is probably not safe to estimate too closely on the amount of exhaust steam given to the heating system. The maximum amount of 44500 pounds per hour was obtained, in this case, by pumping one gallon of water per hour for each square foot of radiation and by pumping city water, in addition to that obtained from the engines. In heating, a less amount of water than this may be circulated even on the coldest day. This is possible, first, because water may be carried at a higher temperature than that stated, and second, because there may be less loss of heat in the conduit, thus giving more heat per gallon of water to the radiation. Again, in estimating for a city water supply, the demands are not very constant and are difficult to estimate. In this one design it was thought that 44500 pounds per hour was a very liberal allowance and could be dropped to 35000 pounds (140000 square feet of radiation), when estimating the amount of radiation supplied by the exhaust steam.

By Fig. 164 it will be seen that the minimum load on the steaming boilers carries through six hours out of the entire twenty-four and that the exhaust steam at this time drops to 22890 pounds per hour, supplying 91560 square feet of radiation. This minimum load is 51 per cent. of the maximum, and 66 per cent. of the amount taken as an average, i. e., 35000. The work done by the economizer is fairly constant, since the amount of economizer surface lost by the steaming boilers under minimum load would be made up by the additional heating boilers thrown into service. *On the basis of 35000 pounds per hour*, the exhaust steam and the stack gases together would heat 170960 square feet and there would be left 13540 square feet ($184500 - 20 \times 1548 - 4 \times 35000$), to be heated by additional boilers. Under minimum load this would be approximately 122500, leaving 62000 square feet to be heated by additional boilers. If one boiler horse-power supplies 160 square feet of radiation, it would require 84 and 387 boiler horse-power respectively to supply the deficiency and the total horse-power needed in each case would be 1632 and 1935. A more satisfactory analysis, how-

ever, is the following which is worked *on the basis of 44500 pounds per hour.*

Let W_s = total number of pounds of steam used in the plant per hour = approximate number of pounds of exhaust steam available for heating the circulating water per hour; W_e = equivalent number of pounds of steam evaporated from and at 212° ; λ = total heat, above 32°, in one pound of dry steam at the boiler pressure; q' = total heat, above 32°, in one pound of feed water entering the boiler; then, if the latent heat of steam at atmospheric pressure = 970.4 B. t. u. we have

$$W_e = \frac{W_s\ (\lambda - q')}{970.4} \tag{123}$$

and the corresponding boiler horse-power needed as steaming boilers will be

$$B_s.\ H.\ P. = \frac{W_e}{34.5} \tag{124}$$

Next the radiation in the district that can be supplied by the exhaust steam is $R_w = 4\ W_s$, and the amount supplied by the economizer is $R_e = 20 \times B.\ H.\ P.$ From which we may obtain the capacity of the heating boilers, as

$$B_w.\ H.\ P. = \frac{Total\ Radiation - 4\ W_s - 20\ B.\ H.\ P.}{160} \tag{125}$$

The total boiler horse-power of the plant is, therefore, the sum of $B_s.\ H.\ P.$ and $B_w.\ H.\ P.$ To obtain Equation 125 for any specific case one must consider the maximum and minimum conditions of the steaming boiler plant. Let W_s (max) = maximum exhaust steam, and W_s (min) = minimum exhaust steam. Then for the two following conditions we have, Case 1, *where the steaming and heating boilers are independent* of each other, the total boiler horse-power installed = $B_s.\ H.\ P.$ + [total radiation — 4 W_s (min) — 20 × $B.\ H.\ P.$ in use] ÷ 160. Also, Case 2, *where a part or all of the steaming boilers are piped for both steaming and water service,* the total boiler horse-power installed = $B_s.\ H.\ P.$ + [total radiation — 4 W_s (max) — 20 × $B.\ H.\ P.$ in use] ÷ 160. It will be noticed that the last term representing the economizer service is simply stated as boiler horse-power and no distinction is made between steaming or heating service. This term is difficult to estimate to an exact figure because it should be the total horse-power in use at any one time, both steaming and heating,

and this can only be obtained by approximation. It makes no difference what service the boiler may be used for, the work of the economizer is practically the same. Probably the most satisfactory way is to substitute the value of $B_s. H. P.$ for $B. H. P.$ in the economizer and get the approximate total horse-power, then if this approximate total horse-power differs very much from that actually needed, other trials may be made and new values for the total horse-power obtained until the equation is satisfied.

Application.—Let W_s = pounds of exhaust steam, λ = 1192 (125 pounds gage pressure), and q' = 28 (feed water at 60°); then when $W_s = 44500$

$$W_e = 53400$$

$$B_s. H. P. = 1548$$

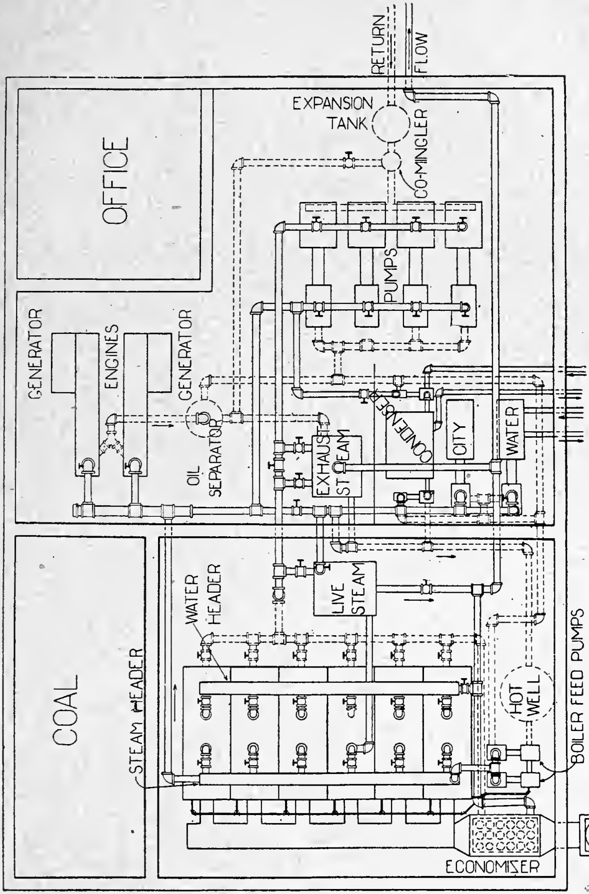
$$B_w. H. P. \text{ Case } 1 = \frac{184500 - 4 \times 22890 - 20 \times 1548}{160} = 387$$

$$B_w. H. P. \text{ Case } 2 = \frac{184500 - 4 \times 44500 - 20 \times 1548}{160} = -153$$

This shows that there is in excess of waste heat in Case 2, making a total boiler horse-power, Case 1, = 1935 and Case 2, = 1548. Investigating Case 1 to see what error was introduced by using 1548 in the economizer, we find approximately 800 horse-power of steam boilers in use, and the total horse-power to be 1187, which is about 360 horse-power on the unsafe side. Substitute again and check results. Case 2 is reasonably close. In any case the most economical size of boiler plant to install in a plant requiring both steaming and heating boilers is one where at least a part, if not all, of the boilers are piped so as to be easily changed from one system to the other. By such an arrangement the capacity may be made the smallest possible. After obtaining the theoretical size of the plant, it would be well to allow a small margin in excess so that one or two boilers may be thrown out of commission for repairs and cleaning without interfering with the working of the plant. Case 2 seems to be the better arrangement. Assuming 1800 total boiler horse-power we might very well put in six 300 *H. P.* boilers arranged in three batteries.

193. Cost of Heating from a Central Station (Direct Firing):—It will be of interest in this connection to estimate approximately the fuel cost in supplying heat by direct firing

to one square foot of hot water radiation per year from the average central station. In doing this make the boiler assumptions the same as Art. 188. Take coal at 13000 B. t. u.



POWER PLANT LAYOUT.

Fig. 169.

per pound, 2000 pounds per ton, and a boiler efficiency of 60 per cent. Water enters the boiler at 155 degrees from the

returns, and is delivered to the mains at 180 degrees. From the value of the coal we have 15600000 B. t. u. per ton given off to the water. This is equivalent to heating 624000 pounds, or 74910 gallons, of water. If one ton of coal costs 3.50 at the plant, we have

$$350 \div 74910 = .0047 \text{ cent}$$

This represents the expense for fuel to reheat one gallon of water, or to supply one square foot of heating surface one hour at an outside temperature of zero degrees. Let the average outside temperature for the eight heating months be 39° (see Art. 63). This gives an average difference between the inside and outside temperatures in any residence of 70 — 39 = 31 degrees, and the equation for the heat loss, Art. 39, reduces to 31 ÷ 70 = .44 of its former value. Now, if it takes one gallon of water per square foot of radiation per hour under maximum conditions, we have for the eight months .44 × 8 × 30 × 24 = 2535 gallons of water heated for each square foot of radiation, at a fuel expense of 2535 × .0047 = 11.9 cents per square foot of radiation for the heating year.

When the plant is working under the best conditions this figure can be reduced. It can be done with boilers of a higher efficiency than that stated, or by using a cheaper coal, both of which are possible in many cases.

194. Cost of Heating from a Central Station. Summary of Tests:—The following tests were conducted upon the Merchants Heating and Lighting Plant, LaFayette, Ind.; one in 1906 and the other in 1908. The plant was changed slightly between the two tests and the radiation carried upon the lines was much increased, although in all essential features the plant was the same. The circulating water was heated by exhaust steam heaters and by heating boilers.

The plant had the following important pieces of apparatus employed in generating or absorbing the heat supply:

BOILERS (Steaming and Heating).

Two 125 *H. P.* Stirling boilers. Total heating surface 2524 sq. ft.

Three 250 *H. P.* Stirling boilers. Total heating surface 7572 sq. ft.

Pressure on steam boilers (gage), 150 lbs.

Pressure on heating boilers (approx.), 60 lbs.

ENGINES

One 450 *H. P.* Hamilton Corliss comp. engine, direct connected to a 300 *K. W.* Western Electric 72-pole alternating current generator 120 *R. P. M.* This engine carried the load of the plant when it was above 50 *K. W.*, which was generally from 5:30 A. M. to 11:30 P. M. When this unit was run, direct current was obtained by passing the alternating current through a motor generator set.

One 125 *H. P.* Westinghouse comp. engine, belted to one 75 *K. W.* 3-phase alternating and two direct current generators, and run at 312 *R. P. M.* This unit was generally run between 11:30 P. M. and 5:30 A. M.

One 250 *H. P.* Westinghouse comp. engine, belt connected to a 200 *K. W.* generator and two smaller machines.

PUMPS.

One centrifugal, two-stage pump, Dayton Hydraulic Co., direct connected to a Bates vertical high speed engine at 300 *R. P. M.*

Two Smith-Vaile horizontal recip. duplex pumps 14 in. × 12 in. × 18 in. Each of the three pumps connected to the return main in such a way as to be able to use any combination at any one time to circulate the water. The centrifugal pump had been in service only one season. It had a capacity about equal to the two reciprocating pumps and under the heaviest service this pump and one of the duplex pumps were run in parallel.

One Smith-Vaile horizontal reciprocating tank pump 6 in. × 4 in. × 6 in. to lift the water of condensation from the exhaust heater to the tank.

One Smith-Vaile horizontal reciprocating make-up pump 6 in. × 4 in. × 6 in. to replace the water that was lost from the system.

Two National horizontal reciprocating boiler feed pumps.

One 9½ in. Westinghouse air pump, to keep up the supply of air through the conduits to the regulator system in the heated buildings.

One Deane vertical deep well pump, to deliver fresh water to the supply tank.

One Baragwanath exhaust steam heater or condenser, having 1000 sq. ft. of heating surface.

PARTIAL SUMMARY OF RESULTS.

		1906	1908
1.	Square feet of radiation	118000	150000
2.	Temperature of circulating water in degrees F., flow main	158.36	164.4
3.	Temperature of circulating water in degrees F., return main	139.9	139.6
4.	Temperature of circulating water in degrees F., after leaving heater	145.6	147.
5.	Temperature of outside air in degrees F.	32.6	37.5
6.	Temperature of stack gases in degrees F., steaming boiler		566.8
7.	Temperature of stack gases in degrees F., heating boiler	562.	656.
8.	Draft in stacks (all boilers average) in inches of water	.689	.595
9.	Heating value of coal in B. t. u. per pound	12800	11565
10.	B. t. u. delivered to steaming boiler per hour by coal	18187000	25833000
11.	B. t. u. delivered to heating boilers per hour by coal	19226000	27917000
12.	B. t. u. delivered to circulating water by heating boilers per hour	11800000	15405000
13.	B. t. u. to be charged to heating boilers (Item 12—Item 15)	7650000	6934000
14.	B. t. u. delivered to circulating water by exhaust steam from the generating engines per hour	3600000	6602000
15.	B. t. u. thrown away during test from pump exhausts and available for heating circulating water	4150000	8471000
16.	B. t. u. available for heating circulating water from all exhaust steam as in normal running (Item 14 + Item 15)	7750000	15073000
17.	Total B. t. u. given to circulating water per hour (Item 13 + Item 16)	15400000	22007000
18.	Gallons of water pumped per hour [Item 17 ÷ (8.33 × Items 2—3)]	100000	108000

19. Gallons of water pumped per square foot of radiation per hour (Item 18 ÷ Item 1)	.85	.70
20. Efficiency of heating boilers (Item 12 ÷ Item 11) approx.	.60	.55
21. Value of the coal in cents per ton of 2000 pounds at the plant	200.	175.
22. Average electrical horse-power	68	141

Note.—The above values are averages and were taken for each entire test. The B. t. u. values were considered satisfactory when approximated to the nearest thousand.

195. Regulation:—The regulation of the heat within the residences is best controlled from the power plant. In most heating plants a schedule is posted at the power house which tells the engineer the necessary temperature of the circulating water to keep the interior of the residences at 70 degrees with any given outside temperature. The Merchants Heating and Lighting Company mentioned above use the following schedule:

Atmosphere	Water	Atmosphere	Water
60 deg.	120 deg.	10 deg.	190 deg.
50 "	140 "	0 "	200 "
40 "	150 "	—10 "	210 "
30 "	160 "	—20 "	220 "
20 "	180 "		

In addition, read the article by Mr. G. E. Chapman, published in the Heating and Ventilating Magazine, August, 1912, page 23, in which he describes the methods used in regulating the Oak Park, Ill., plant.

In some heating plants the regulation is by means of air carried from the compressor at the power house through a main running parallel with the water mains in the conduits, and branching to each building where it is used under a pressure of 15 pounds to operate thermostats, which in turn control the water inlets to the radiators. A closer regulation is obtained in the latter system than in the former, but it is needless to say that the thermostats require careful adjustments and frequent inspections and repairs.

Diaphragms or *chokes* having different sized orifices may be placed on the return main from each building to regulate the supply. Those buildings nearest to the power plant

have the advantage of a greater differential pressure than those farther away, hence should have smaller diaphragms. By increasing the resistance in the return line from any building the water circulates more slowly and has time to give off more heat to the rooms. With a high temperature of the water and a careful adjustment of the diaphragms it is possible to have the amount of water circulated per square foot of radiation reduced much below one gallon per square foot per hour.

STEAM SYSTEMS.

196. Heating by steam from a central station, compared with hot water heating, is a very simple process. The power plant equipment is composed of a few inexpensive parts, the operation of which is very simple and easily explained. These parts have but few points that require rational design. Because of the simplicity and the similarity to the preceding discussion on hot water systems, the work on steam systems will be very brief. All questions referring to the construction of the conduit, the supporting of the pipes, the provision for contraction and expansion, and the draining of the pipes and conduits, are common to both hot water and steam systems and are discussed in Arts. 160 and 161. A large part of the work referring directly to district hot water heating applies with almost equal force to steam heating. This part of the work, therefore, will deal with such parts of the power plant equipment as differ from those of the hot water system.

Centralized steam heating may be classified under two general heads, high pressure and low pressure, referring to the pressures carried in the transmission lines. Ordinarily steam is generated at high pressure at the boiler, 60 to 150 pounds gage, and reduced for line service to pressures varying from 5 to 30 pounds gage, with a still further reduction at the building to pressures varying from 0 to 10 pounds gage for use in radiators and coils. Where exhaust steam is used in the main, the pressure is not permitted to go higher than 10 pounds gage, because of the back pressure on the engine or turbine. Where exhaust steam is not used, the pressures may be carried as high as desired, thus allowing for a greater pressure drop in the line and a corresponding reduction in pipe sizes. (See Central Station Heating in De-

troit. Power, May 7, 1918). In large plants the necessity for high pressures and small pipes is apparent. Even in lines carrying exhaust steam, high pressure *feeders* or *boosters* are frequently run parallel to the heating main and at stated points connect to the heating main through pressure reducing valves. *Vacuum returns* may be applied to central station work the same as to isolated plants. (See Art. 159—Returning the water to the power plant. Also, Chap. IX).

The principles involved in the power plant end of a steam heating system may be represented by Fig. 170. It will be seen that the exhaust steam from the engines or turbines has four possible outlets. Passing through the oil separator, which removes a large part of the entrained oil, part of the exhaust steam is turned into the heater for use in heating the boiler feed water. The rest of the steam passes on into the heating system. If there is more exhaust steam than is necessary to supply the heating system, the balance may go to the atmosphere through the back pressure valve. When the heating system is not in use, as would be the case in the four warm months of the year, the exhaust steam may be passed into the condenser.

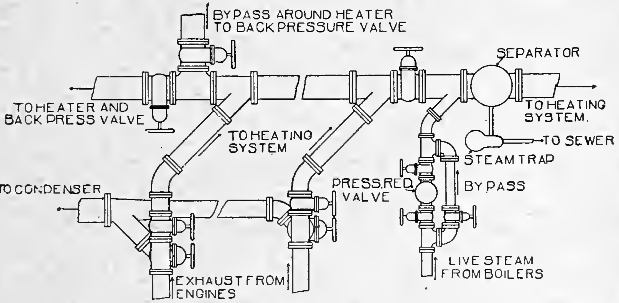


Fig. 170.

It is very evident, from what has been said before, that it would not be economical to condense the steam in a condenser as long as there is a possibility of using it in the heating system. The increased gain in efficiency, when condensing the exhaust steam under vacuum, is very small compared to the gain when this same steam is used for heating

purposes. It would be also very poor economy to use any live steam for heating when there is any exhaust steam wasted. When the amount of exhaust steam is insufficient, live steam is admitted through a pressure reducing valve.

197. Drop in Pressure and the Diameter of the Mains:— The flow of steam in a pipe follows the same general law as the flow of water. The loss of head may be represented by the well known equation,

$$h_f = \frac{2\,\phi\, l\, v^2}{g\, d} \tag{126}$$

where h_f = loss of head in feet, ϕ = coefficient of friction, v = velocity in feet per second, l = length of pipe in feet, d = diameter of the pipe in feet and $g = 32.2$. Substitute, $h_f = 144\ p \div D$, where p = drop in pressure in pounds and D = density of the steam, and find

$$p = \frac{2\,\phi\, l\, v^2\, D}{144\, g\, d} \tag{127}$$

The coefficient of friction is found to vary with the velocity of the steam and with the diameter of the pipe. Prof. Unwin found that for velocities of 100 feet per second (good practice for transmission lines), it could be expressed as follows, where c is a constant to be found by experiment,

$$\phi = c\left(1 + \frac{3}{10\, d}\right)$$

which when substituted in Equation 127, gives

$$p = \frac{l\, v^2\, D\, c}{72\, g\, d}\left(1 + \frac{3}{10\, d}\right) \tag{128}$$

Let W = pounds of steam passing per minute and d_1 = diameter of pipe in inches, then

$$p = \frac{1}{20.663}\left(1 + \frac{3.6}{d_1}\right)\frac{W^2\, l\, c}{d_1^{5}\, D} \tag{129}$$

The recommended value of the constant c for steam is .0027. From this equation we may obtain any one of the five terms, d_1, W, p, l or D, if the other four are known or assumed. In the greatest number of practical problems the item desired is the *diameter*, d_1, and conditions must give the pounds of steam to be conveyed per minute, the pressure drop allowable for its transmission, the length of transmission pipe and

the steam pressure (or density). In a comparatively few problems W or p may be required and the other four items given.

APPLICATIONS BY THE USE OF EQUATION 129.

APPLICATION 1.—A steam power main is to be designed to deliver 8400 pounds of steam per hour, at 100 lbs. gage pressure, through a distance of 1000 feet of straight pipe. What will be the *diameter* if the allowable pressure drop for this 1000 foot run is first, 1 lb.; second, ½ lb.; third, 10 lbs.?

SOLUTION.—8400 pounds per hour = 140 pounds per minute. At 100 lbs. gage pressure the density of the steam is .258 and $p = \frac{1}{20.663}\left(1 + \frac{3.6}{d_1}\right)\frac{140 \times 140 \times 1000 \times .0027}{d_1^5 \times .258}$;

Reducing $p = \frac{9938}{d_1^5} + \frac{35768}{d_1^6}$ in which,

when $p = 1$, $d_1 = 6.9''$ (area 37.40); 7″ main required.
" $p = ½$, $d_1 = 7.8''$ (area 47.78); 8″ main required.
" $p = 10$, $d_1 = 4.5''$ (area 15.90); 4½″ main required.

APPLICATION 2.—A 4-inch steam heating main 700 feet long is receiving steam at 15 lbs. gage pressure and delivering it at a pressure 1% lower. What *quantity* of steam is being delivered? What *quantity* will be delivered if a drop of three pounds is allowed?

SOLUTION.—$.15 = \left(.0484 + \frac{.1742}{4}.\right)\frac{W^2 \times 700 \times .0027}{1024 \times .072}$;

$.15 = \frac{.0919\,W^2 \times 700 \times .0027}{1024 \times .072}$; $W = \sqrt{\frac{11.06}{.1736}} = 7.9$ lbs. per min. Since with everything else constant the quantity of steam varies as the square root of the pressure drop, for three pounds drop $7.9 : \sqrt{.15}$ as $W_1 : \sqrt{3}$, whence $W_1 = 4.5 \times 7.9 = 35.6$ lbs. per min.

APPLICATION 3.—The equivalent length of a 4-inch high pressure steam main is to be 1600 feet and it will be expected to deliver 9000 pounds of steam per hour when the pressure is 150 lbs. gage. What *pressure drop* will be experienced when delivering this amount?

SOLUTION.—9000 pounds per hour = 150 pounds per minute. At 150 lbs. gage pressure the steam density is .3635, then $p = \left(.0484 + \frac{.1742}{4} \right) \frac{150 \times 150 \times 1600 \times .0027}{1024 \times .3635} =$ 23.8 lbs.

APPLICATIONS BY TABLE.

To avoid the time and labor required in solving Equation 129, tables have been compiled and curves plotted. None of these time saving efforts, however, have produced a working scheme which is perfectly general, as all have at least two of the five variables constant. Thus, Table 39, Appendix, was compiled from Equation 129, upon the basis of a constant pressure drop, $p = 1$ pound, and a constant pressure of 100 lbs. absolute in the pipe. *When these two conditions obtain, values may be read directly from the table, but when the pressure or the pressure drop differs from these, corrective calculations must be applied to the tabular values.*

As may be observed from the equation, the drop in pressure is proportional to the square of the pounds of steam flowing per minute (other items constant) and the amount delivered at *any other pressure drop* than that of the table, (1 pound) will be found by multiplying the reading from the table by the square root of the desired pressure drop in pounds. Also, since the *weight* of steam moved at the same velocity under any other absolute pressure is approximately proportional to the absolute pressures (other items constant), the amount delivered at *any other pressure* will be found by multiplying the reading from the table by the square root of the ratio of the absolute pressures. The use of the table will be made clear by the following checks of Applications 1, 2 and 3 above.

Check 1. Since the pressure in Application 1 is 100 lbs. gage and the table is calculated for 100 lbs. absolute, quantities of steam, before insertion into table, must be multiplied by $\sqrt{\frac{100}{115}}$. Hence the check of that part of Application 1 having one pound drop is as follows:

$140 \times \sqrt{\frac{100}{115}} = 130$ lbs. corrected steam. In column

under 1000 feet, find by interpolation that 130 lbs. per minute corresponds to a diameter of 6.9; therefore a 7-inch main is required. Before being ready to refer the quantity of steam to the table for checking that part having .5 lb. pressure drop, it is necessary to apply corrections for both pressure and pressure drop, thus,

$140 \times \sqrt{\frac{100}{115}} \times \sqrt{\frac{1}{.5}} = 185$ lbs. corrected steam. Under 1000 feet, find by interpolation, that 185 lbs. per minute corresponds to a diameter of 7.8 = 8-inch main.

Similarly the 10 lb. drop is corrected by

$140 \times \sqrt{\frac{100}{115}} \times \sqrt{\frac{1}{10}} = 41.3$ lbs. corrected steam. Under 1000 feet, find by interpolation, that 41.3 lbs, per minute corresponds to a 4½-inch main.

Check 2. In Table 39, under 700 feet, at 4-inch diameter, the capacity of the main is given as 36.7 lbs. *for the conditions* of 100 lbs. pressure absolute and 1-lb. drop in pressure. The corrective factor for pressures is evidently $\sqrt{\frac{30}{100}}$. Likewise the corrective factor for pressure drop is $\sqrt{\frac{.15}{1}}$. From these conditions the capacity is $36.7 \times \sqrt{\frac{30}{100}} \times \sqrt{\frac{.15}{1}} =$ 7.8 lbs. per min. For the 3-lb. drop, the corrective calculations are $36.7 \times \sqrt{\frac{30}{100}} \times \sqrt{\frac{3}{1}} = 35.5$ lbs per min.

Check 3. In Table 39, under 1600 feet, at 4-inch diameter the capacity of the main is given as 24.4 lbs. for conditions of 100 lbs. pressure absolute and 1-lb. drop in pressure. The corrective factor for pressures increases this as follows:

$24.4 \times \sqrt{\frac{165}{100}} = 31.3$ lbs. per minute, being the capacity of the main at the *problem pressure,* but at the *pressure drop of the table,* 1-lb. Since capacities are proportional to the square root of the pressure drops,

$31.3 : 150$ as $\sqrt{1} : \sqrt{p}$ whence $p = 23$ pounds drop.

It will be seen that the corrections, necessary because of the two items assumed constant, are always made to affect the *quantity of steam involved*, and upon analysis the following *general directions* for finding either a required W, as in Application 2, or a tabular W, as in Application 1, will be found to hold.

$$\text{Required } W \times \sqrt{\frac{\text{pressure of table}}{\text{pressure required}}} \times \sqrt{\frac{\text{drop in table}}{\text{drop required}}} = \text{Tabular } W$$

$$\text{Actual Steam} = \text{Steam from Table} \times \sqrt{\frac{\text{given pressure}}{\text{pressure basis of table}}} \times \sqrt{\frac{\text{given drop}}{\text{drop in table}}}$$

$$\text{Steam to be taken in Table} = \text{Actual Steam} \times \sqrt{\frac{\text{pressure basis of table}}{\text{given pressure}}} \times \sqrt{\frac{\text{drop in table}}{\text{given drop}}}$$

198. Dripping the Condensation from the Mains:—The condensation of the steam which takes place in the conduit mains, should be dripped to the sewer or the return at certain specified points, through some form of steam trap. These traps should be kept in first-class condition. They should be inspected every seven to ten days. No pipe should be drilled and tapped for this water drip. The only satisfactory way is to cut the pipe and insert a tee with the branch looking downward and leading to the trap. The sizes of the traps and the distances between them can only be determined when the pounds of condensation per running foot of pipe can be estimated.

199. Adaptation to Private Plants:—District steam heating systems may be adapted to private hot water plants by the use of a "transformer." This in principle is a hot water tube heater which takes the place of the hot water heater of the system. It may also be adapted to warm air systems by putting the steam through indirect coils and taking the air supply from over the coils.

200. General Application of the Typical Design:—The following brief applications are meant to be suggestive of

the method only, and the discussions of the various points are omitted.

Square feet of radiation in the district.—

$R_s = 184500 \times 170 \div 255 = 123000$ square feet.

Amount of heat needed in the district to supply the radiation for one hour in zero weather.—

Total heat per hour $= 123000 \times 255 = 31365000$ B. t. u.

Amount of heat necessary at the power plant to supply the radiation for one hour in zero weather.—Assuming 15 per cent. heat loss in the conduit (this is slightly less than that allowed for the hot water two-pipe system, 20 per cent.), we have $31365000 \div .85 = 36900000$ B. t. u. per hour.

Total exhaust steam available for heating purposes.—

W_s (max.) $= (23100 + 8680) \times 1.15 = 36547$ pounds per hour.

W_s (min.) $= (1490 + 8680) \times 1.15 = 11696$ pounds per hour.

Total B. t. u. available from exhaust steam per hour for heating.—Let the average pressure in the line be 5 pounds gage and let the water of condensation leave the indirect coils in the residences at 140 degrees. We then have from one pound of exhaust steam, by Equation 97,

B. t. u. $= .85 \times 960 + 196 - (140 - 32) = 904$

Assuming this to be 900 B. t. u. per pound, the total available heat from the exhaust steam for use in the heating system is, maximum total $= 32892300$ B. t. u. and the minimum total, $= 10526400$ B. t. u.

Square feet of steam radiation that can be supplied by one pound of exhaust steam at 5 pounds gage.—

$R_s = 900 \div (255 \div .85) = 3.$

Total B. t. u. to be supplied by live steam.—

B. t. u. (max. load) $= 36900000 - 32892300 = 4007700$ B. t. u.

B. t. u. (min. load) $= 36900000 - 10526400 = 26373600$ B. t. u.

Total pounds of live steam necessary to supplement the exhaust steam.—Let the steam be generated in the boiler at 125 pounds gage. With feed water at 60 degrees

Max. load $= 4007700 \div 1164 = 3444$ pounds.

Min. load $= 26373600 \div 1164 = 22661$ pounds.

Boiler horse-power needed for the steam power units.—As in Arts. 189 and 192.

$B_s.\ H.\ P.$ (max.) $= 36547 \times 1.2 \div 34.5 = 1271.$

$B_s.\ H.\ P.$ (min.) $= 11696 \times 1.2 \div 34.5 = 407.$

Total boiler horse-power needed in the plant.—Maximum load. $B.\ H.\ P.$ (total) $= 1271 + (3444 \times 1.2 \div 34.5) = 1391.$

It will be noticed that this total horse-power is 157 horse-power less than the corresponding Case 2 in Art. 192. This is accounted for by the fact that no steam is used up in work in the circulating pumps, also that the conditions of steam generation and circulation are slightly different. 1500 boiler horse-power would probably be installed in this case.

Size of conduit mains.—Let it be required to find the diameters of the main system in Fig. 166 at the important points shown. Art. 169 gives the length of the mains in each part. Allow .3 pound of steam for each square foot of steam radiation per hour (this will no doubt be sufficient to supply the radiation and conduit losses). Try first, that part of the line between the power plant and A, with an average steam pressure in the lines of about 5 pounds gage and a drop in pressure of 1½ ounces per each 100 feet of run (approximately 5 pounds per mile). 25200 pounds per hour gives $W = 420$. The length of this part of the line is 200 feet and the drop is 3 ounces, or .19 pound.

$$W \text{ (table)} = \frac{420}{\sqrt{.19}} \times \sqrt{\frac{100}{20}} = 2158 \text{ pounds}$$

which gives a 15 inch pipe.

Following out the same reasoning for all parts of the line, we have

TABLE XXXIX.

	P P to A	A to B	B to C	C to D	D to E
Distance between points	200	500	1500	1500	500
Radiation supplied, sq. ft.	84000	57000	34000	19000	8000
Pressure-drop in pounds = p	.19	.47	1.4	1.4	.47
Diam. of pipe in inches, by table	15	13	11	9	5

In general practice, these values would probably be taken 16, 14, 12, 10 and 6 inches respectively. Look up Table 39, Appendix, and check the above figures.

CHAPTER XIV.

TEMPERATURE CONTROL IN HEATING SYSTEMS.

201. From tests that have been conducted on heating systems, it has been shown that there is less loss of heat from buildings equipped with automatic temperature control, than from buildings where there is no such control. A uniform temperature within the building is desirable from all points of view. Where heating systems are operated, even under the best conditions without such control, the efficiency of the system would be increased by its application. No definite statement can be made for the amount of heat saved, but it is safe to say that it is between 5 and 20 per cent. A building uniformly heated during the entire time, requires less heat than if a certain part or all of the building were occasionally allowed to cool off. When a building falls below normal temperature it requires an extra amount of heat to bring it up to normal, and when the inside temperature rises above the normal, it is usually lowered by opening windows and doors to enable the heat to leave rapidly. High inside temperatures also cause a correspondingly increased radiation loss. Fluctuations of temperature, therefore, are not only undesirable for the occupants, but they are very expensive as well.

202. Principles of the System:—Temperature control may be divided into two general classifications,—small plants and large plants. The *control for small plants*, i. e., such plants as contain very few heating units, is accomplished by regulating the drafts by special dampers at the combustion chamber. This method controls merely the process of combustion and has no especial connection with individual registers or radiators, it being assumed that a rise or fall of temperature in one room is followed by a corresponding effect in all the other rooms. This method assumes that all the heating units are very accurately proportioned to the respective rooms. The dampers are operated by a motor device which in turn is driven by springs or weights and controlled by a thermostat and electric batteries. This system of regulation may be applied to any system of heat.

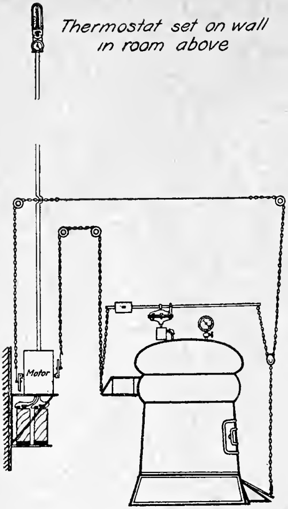


Fig. 171.

Fig. 171 shows a typical application to a small steam boiler plant. Furnace systems require thermostatic control only between the room and the dampers; closed hot water, steam and vapor systems, however, should have additional regulation from the pressure within the boiler to the draft. Occasionally in the morning the pressure in these systems may become excessive before the house is heated enough for the thermostat to act. With such dual regulation no hot water heater or steam boiler would be forced to a dangerous pressure. Fig. 172 represents a standard form of the thermostat supplied by the Andrews Heating Co. and the Minneapolis Regulator Co. The complete regulator has in addition to this, two cells of open circuit battery and a motor box (Fig. 171), all of which illustrate very well the thermostatic damper control.

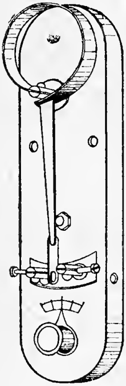


Fig. 172.

The thermostat operates by a differential expansion of the two different metals composing the spring at the top. Any change in temperature causes one of the metals to expand or contract more rapidly than the other and gives a vibrating movement to the projecting arm. This is connected with the batteries and with the motor in such a way that when the pointer closes the contact with either one of the contact posts, a pair of magnets in the motor causes a crank arm to rotate through 180 degrees. A flexible connection between this crank and the damper causes the damper to open or close. A change in temperature in the opposite direction makes contact with the other post and reverses the movement of the

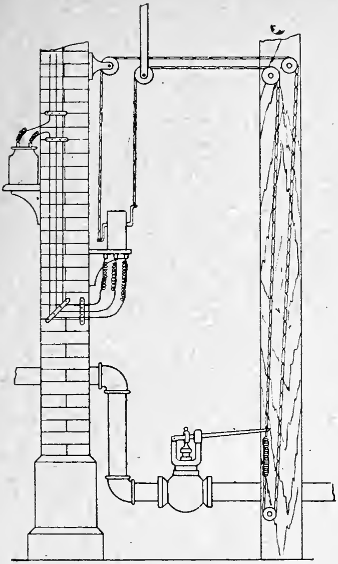


Fig. 173.

crank and damper. The movement of the arm between the contacts is very small thus making the thermostat very sensitive. No work is required of the battery except that necessary to release the motor.

Occasionally it is desirable to connect small heating plants having only one thermostat in control, to a central station system. Fig. 173 shows how the supply of heat may be controlled by the above method.

Temperature control in large plants, i. e., those plants having a large number of heating units, is much more complicated. The following discussion will apply especially to hot water and steam systems, and will be additional to the control at

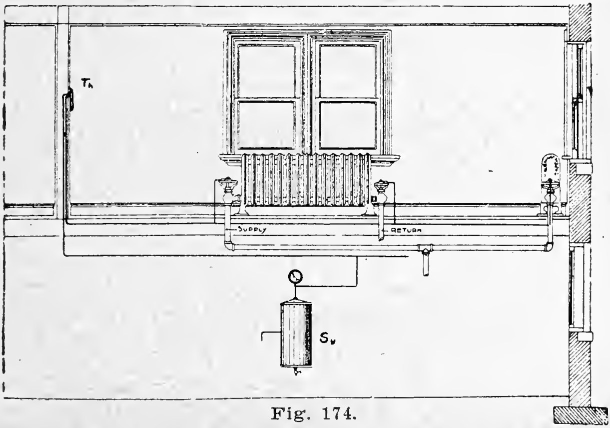


Fig. 174.

the heater and boiler as discussed under small plants. Fig. 174 shows a typical layout of such a system. Compressed air at 15 pounds gage is maintained in cylinder, S_u, which is located in some convenient place for the attendant. This air is carried to the thermostat, T_h, on one of the protected walls in the room. Here it passes through a controlling valve and is then led to the regulating valve on the radiator where it acts on the top of a rubber diaphragm as shown in Fig. 175 to close the valve and to cut off the supply. When the room cools off, the controlling valve at T_h cuts off the supply and opens the radiator air line to the atmosphere. This removes the air pressure from above the diaphragm and permits the stem of the valve to lift. On the opening of the valve the steam or water again enters the radiator and the cycle is completed.

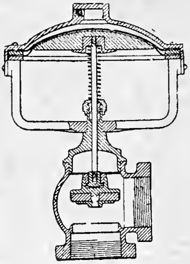


Fig. 175.

Fig. 139 shows the application of thermostatic control to blower work. In this system (single duct system) the thermostat B and the mixing dampers are located at the plenum chamber. The same general arrangement may be applied to the double duct system, with the dampers in the wall at the base of the vertical duct leading to the room.

203. Some of the Important Points in the Installation:— Each radiator has its own regulating valve. All rooms having three radiators or less are provided with one thermostat. Large rooms having four or more radiators have two or more thermostats with not more than three radiators to the thermostat. Where other motive power is not available for the air supply, a hydraulic compressor is used. This compressor automatically maintains the air pressure at 15 pounds gage in the steel supply tank. The main air trunk lines are galvanized iron, ⅜- and ½-inch in diameter, and are tested under a pressure of 25 pounds gage. All branch pipes are ¼- and ⅛-inch galvanized iron. Fittings on the ⅛-inch pipes are usually brass. Where flexible connections are made, this is sometimes done by armoured lead piping. Thermostats are usually provided with metallic covers and are finished to correspond with the hardware of the respective rooms. Each thermostat is provided with a thermom-

eter and a scale for making adjustments. Each radiator is provided with a union diaphragm valve having a specially prepared rubber diaphragm with felt protection. This valve replaces the ordinary radiator valve. One of these valves is used on the end of each hot water radiator, one on each one-pipe steam radiator and two on each two-pipe low pressure steam radiators. This last condition does not hold for two-pipe steam radiators with mechanical vacuum returns, in which case patented specialties are applied by the vacuum company. In such cases the supply to the radiator only is controlled. In any first class system of control, the temperature of the room may easily be kept within a maximum fluctuation of three degrees.

204. Some Special Designs of Apparatus:—All temperature control work is solicited by specialty companies, each having a patented system. In the essential features these systems all agree with the foregoing general statements. The chief difference is in the principle upon which the thermostat, Th, operates.

Fig. 176 shows sections through the intermediate and positive thermostats manufactured by the Johnson Service Company, Milwaukee. The interior workings of the thermostats are as follows: *Intermediate.*—Air enters at A from the supply tank, passes into chamber B and escapes at port C. If thermostatic strip T expands inward to close C, the air pressure collects in B and presses down port valve V, thus opening port E, letting air through into F and out at G to close the damper. When T expands outward, pressure at B is relieved and V is forced back by a spring, closing E. Air in F reacts against the diaphragm and escapes through hollow valve V at H, permitting the damper to open. *Positive.*—Air enters at A, passes into chamber B and escapes at C. If thermostatic strip T expands inward to close C, air pressure collects in B, forces out the knuckle joint K and operates the three-way valve V, thus shutting port E and opening port F, letting air escape and radiator valve open. When T expands outward, pressure at B is relieved, knuckle joint K returns, pulling V outward, thus shutting port F, opening E, letting air escape through G and shutting off radiator valve.

The real thermostat is the spring T. This is composed of steel and brass strips brazed together. Because of a

higher coefficient of expansion in the brass than in the steel, a change in the room temperature causes the spring to move toward or away from the seat *C*. *T* is adjustable for any desired room temperature. The intermediate thermostat is used on indirect heating where mixing dampers are employed and where an intermediate position of the valve is necessary. The positive thermostat is used on direct radiators and coils where a full open or full closed movement of the valve is desired.

INTERMEDIATE POSITIVE

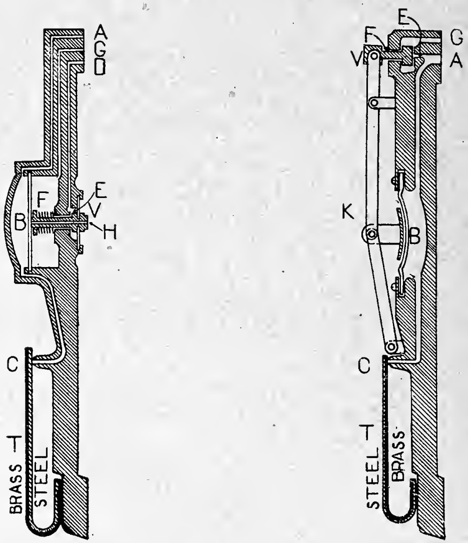


Fig. 176.

Fig. 177 shows a section through pattern *K* thermostat, manufactured by the Powers Regulator Co., Chicago. This thermostat consists of a frame carrying two corrugated disks, brazed together at the circumference and containing a volatile liquid having a boiling point at about 50 degrees F. At a temperature of about 70 degrees, the vapor within the disks has a pressure of about 6 pounds to the square inch. This pressure varies with every change of tempera-

ture and produces variations in the total thickness of the center of the disks.

The compressed air enters at *H* and passes into chamber *N* through the controlling valve *J*, which is normally held to its seat by a coil spring under cap *P*. Within the flange *M* is located an escape valve *L* upon which the point of the supply valve *J* rests. Valve *L* tends to remain open when permitted by reason of the spring underneath the cap. When the temperature rises sufficiently to cause the disks to increase in thickness and move the flange *M*, the first action is to seat the escape valve *L*, its spring being weaker than that above *J*. If the expansive motion is continued after

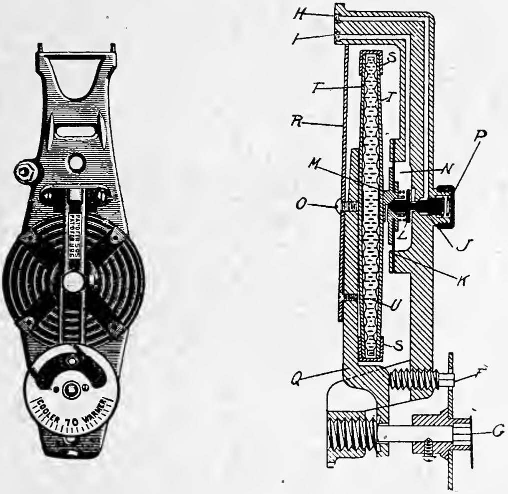


Fig. 177.

valve *L* is seated, the valve *J* is then lifted from its seat and compressed air flows into the chamber *N*. As the air accumulates in chamber *N*, it exerts a pressure upon the elastic diaphragm *K* in opposition to the expansive force of the disk. So, whenever there is sufficient pressure in *N* to balance the power exerted by the disks, the valve *J* returns to its seat and no more air is permitted to pass through. If the temperature falls, the pressure within the disks becomes less, the disks draw together and the over-balancing

air pressure in N reverses the movement of the flange M and permits the escape valve L under the influence of its spring to rise from its seat, whereupon a portion of the air in N is discharged until the pressure in N becomes equal to the diminished pressure from the disks. Thus the pressure of the air in N is maintained always in direct proportion to the expansive power (temperature) of the disks. Port I connects with chamber N and leads to the diaphragm valve.

This thermostatic valve controls the regulator valve by a graduated movement and is used on the dampers for blower work. Another form with maximum movement only is designed for steam systems.

Fig. 178 shows the positive and graduated thermostats as manufactured by the National Regulator company, Chicago. The thermostatic element in these thermostats is the vulcanized rubber tube A, which changes its length with the varying room temperatures and causes the valve O to open or close the port G, thus controlling the supply of air to

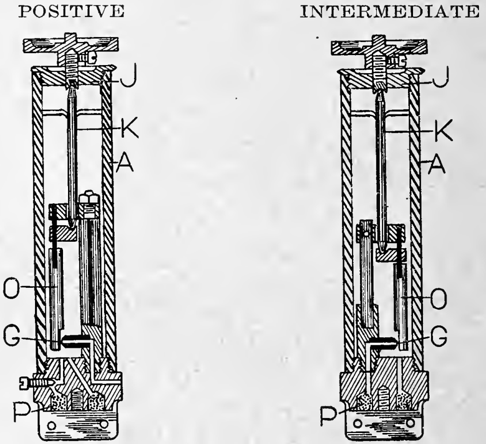


Fig. 178.

and from the radiator valve or the regulating damper. In the positive thermostat air enters the tube from the supply through the filter and restricted passage P. From the interior of the tube the air leaves through the middle orifice and enters the pipe leading to the radiator valve. If the

room temperature is above the normal, port G closes and the air pressure collects in the tube, thus creating a pressure in the line leading to the radiator valve and closing it. If the room temperature falls below the normal, port G opens, air is exhausted from the tube to the atmosphere, the pressure on the radiator valve is released and the valve opens. The intermediate thermostat differs from the positive thermostat in having but one air line. Room temperatures below the normal contact tube A, open port G, and exhaust the air to the atmosphere. With this release in pressure in the pipe at P the regulating damper is turned to admit more warm air into the room. With the room temperature above the normal, tube A expands, port G closes, pressure in pipe P increases and the regulating damper is turned so as to admit a lower temperature of air in the room. By means of this a graduated movement of the damper is obtained.

CHAPTER XV.

ELECTRICAL HEATING.

In the present state of the heating business it seems almost unnecessary to discuss electrical heating, in any serious way, in connection with steam power plants. The reasons will be seen in the following brief discussion. Electrical heating can appeal to the public only from the standpoint of convenience, since a comparison of economies between steam, hot water or warm air heating on one hand, and electrical heating on the other, is wholly against the latter. Its application to the process of heating will find its greatest economy in connection with water power plants where the combustion of fuel is eliminated from the proposition. This discussion will not bear in any way upon the water power generator.

205. Equations Employed in Electrical Heating Design:—

1 H. P. = 746 watts.

1 H. P. = 33000 ft. lbs. per min. = 1980000 ft. lbs. per hr.

1 B. t. u. = 778 ft. lbs.

1 H. P. hr. = 1980000 ÷ 778 = 2545 B. t. u. per hr.

1 H. P. hr. = 746 watt hours = 2545 B. t. u. per hr.

1 watt hr. = 3.412 B. t. u. per hr.

1 watt hr. = 3.412 ÷ 170 = .02 sq. ft. of hot water rad.

1 watt hr. = 3.412 ÷ 255 = .0134 sq. ft. of steam rad.

1 kilo-watt hr. = 20.1 sq. ft. of hot water rad. (130)

1 kilo-watt hr. = 13.4 sq. ft. of steam rad. (131)

206. Comparison between Electrical Heating and Hot Water and Steam Heating:—The loss in transmitting electricity from the generators through the switchboard to the radiators may be small or large, depending upon the conditions of wiring, the current transmitted and the pressure on the line. In all probability it would equal or exceed the transmission losses in hot water or steam lines. Assuming these losses to be the same, a fair comparison may be made in the cost of heating by the various methods. The operating efficiency of an electric heater is 100 per cent., since all the current that is passed into the heater is dissipated in

the form of heat and no other losses are experienced. This is not true of steam systems where the water of condensation is thrown away at fairly high temperatures. Where electricity or steam is generated and distributed all in the same building, there is no line loss to be accounted for, since all of this heat goes to heating the building and counts as additional radiation.

Equations 130 and 131 show the theoretical relation existing between electrical heating and hot water and steam heating compared at the power plant. The following discussion is based, therefore, upon the assumption that 1 kilo-watt hour, in an electric radiator, will give off the same amount of heat as 20.1 and 13.4 square feet of hot water and steam radiation respectively. With coal having 13000 B. t. u. per pound and a furnace efficiency of 60 per cent., it will require $3412 \div 7800 = .44$ pound of coal per hour. If coal costs \$2.00 per ton of 2000 pounds, there will be an actual fuel expense of .044 cent. On the other hand, assuming the combined mechanical efficiency of an engine or turbo-generator set to be 90 per cent., the heat from the steam that is turned into electrical energy per hour is $1000 \div .90 = 1111$ watts, for each kilo-watt delivered. Now if this unit has 15 per cent. thermal efficiency, we have the initial heat in the steam equivalent to $1111 \div .15 = 7400$ watt hours. From this obtain $7400 \times 3.412 = 25249$ B. t. u. per hour; or, $25249 \div 7800 = 3.2$ pounds of coal per hour. This, at the same rate as shown above, would be worth .32 cent. Comparing, the electrical generation actually costs 7.2 times as much as the other. This comparison has dealt with the fuel costs at the plant and has not taken into account the depreciation, labor costs, etc., the object being to show relative efficiencies only.

Another way of looking at this subject is as follows: a fairly large turbo-generator set (say 500 K. W.) will deliver 1 kilo-watt hour to the switchboard on 20 pounds of steam. With 10 per cent. additional steam for auxiliary units, this amounts to 22 pounds of steam per kilo-watt hour at the switchboard. One pound of steam generated in a plant of this kind with the above efficiencies and value of coal, also with a steam pressure of 150 pounds and a good feed water heater, will give to each pound of steam approximately 1000 B. t. u. This makes 22000 B. t. u. or 2.8 pounds of coal re-

quired to each kilo-watt output. This is about 10 per cent. less than the above figures.

The ratio of 7 to 1, as shown in the above efficiencies, does not seem to hold good in the selling price to the consumer. In round numbers, district steam and hot water heating systems supply 25000 B. t. u. to the consumer for one cent. The cost for electrical energy to the consumer is between 6 and 7 cents per kilo-watt. This gives 3412 ÷ 6.5 = 525 B. t. u. for one cent. Comparing with the above, gives a ratio of 48 to 1.

207. The Probable Future of Electrical Heating:—Because of the low efficiency of electrical heating as compared with other methods of heating, it is very probable that it will not replace the other methods in sections of the country having severe or changeable climate, except in so far as the convenience of the user is the principal thing sought for, and the expense of operating a minor consideration.

On the other hand, in limited sections of the country where conditions are favorable (sections of the Pacific Coast for example), heating by electricity is rapidly increasing. Hydro-electric power at a low rate and a climate that requires only a small amount of heat in the buildings, morning and evening, make electric heating an actual economy. In such a climate the cumbersome coal and oil burning steam and water heating systems would be inappropriate, while the starting and the banking of fires when not needed would be a wasteful process.

While in most cases electricity will be barred from the usual heating and laundry processes, it will continue to be increasingly used in those household economies where temperatures are needed above 250°, such as percolators, grills, toasters, broilers, plate warmers, ranges and ovens. Electric ovens in bakeries are being employed because they occupy much less space than the brick oven of same capacity, are light in weight, are more cleanly, can be regulated more easily and use the heat generated much more effectively.

CHAPTER XVI.

REFRIGERATION.

DESCRIPTION OF SYSTEMS AND APPARATUS.

208. General Divisions of the Subject:—The rapidly increasing demand for the cold storage of food products, the production of artificial ice and the cooling of buildings have developed for the heating engineer a broad and inviting field, namely, refrigeration. A municipal electric or pumping station with a district heating plant to utilize the exhaust steam in winter and a refrigeration plant to utilize the same in summer furnishes a unique opportunity for economic engineering. One application of the above principle where a 10-ton ice plant of the absorption type was so operated in a town of 3500 population and earned a dividend of 13 per cent. on the investment, is proof, if any is needed, that the field is an intensely practical one.

As in heating systems there must be sources of heat, circulating mediums, distributing systems and delivering systems whereby the carriers give up their heat at the proper places in the circuits, so in refrigerating systems there must be sources of minus heat or of heat abstraction, circulating mediums, distributing systems and receiving systems whereby the carriers take up heat at the proper places in the circuits from articles or rooms that are being cooled. The carriers (circulating mediums), and the receiving and transmitting of the heat to and from them present no special difficulties or great diversity of practice, but in the methods of producing and maintaining the sources of minus heat there are considerable differences and numerous methods.

209. Refrigerating Systems may be divided into two groups, those producing cold by more or less chemical action between ingredients upon mixing, called *chemical systems,* and those producing cold by the evaporation of a liquefied gas or the expansion of a compressed gas, called mechanical systems. Chemical systems are used only occasionally in commercial work, but are frequently found in small sized plants for domestic purposes. Low first cost and convenience of

handling are the principal advantages. This division includes the simple melting of ice and the mixing of ice and salt for temperatures as low as 0 to —5 degrees. The latter is much used in domestic processes for the production of table ices. etc. Other ingredients used in the mixtures with the corresponding temperature drops which may be expected are given in Table 58, Appendix. The chemical method of producing cold is occasionally used to maintain low temperatures in storage rooms while repairs are being made upon the regular machinery. The chemical methods of cooling are so simple in principle that they will not be discussed further in this work. Mechanical systems include all the practical methods of commercial refrigeration. These are, the *vacuum system*, the *cold air system*, the *compression system* and the *absorption system.*

210. Vacuum Systems:—This system was formerly of some importance but of late years has given place to other and more efficient methods. Fig. 179 shows a vacuum system in diagram. If a spray of water or brine is injected into a chamber that contains pans of sulphuric acid and is kept at a partial vacuum of one or two ounces, the acid absorbs the water vapor from the spray, thus assisting in maintaining the vacuum and lowering the temperature of the remainder of the spray. The vaporization of the part that is absorbed by the acid requires heat. This heat is taken from the liquid of the spray that is not absorbed, consequently the temperature of the remaining liquid is lowered. In a system of this kind a temperature of 32 degrees may easily be obtained. The water or brine after cooling is then circulated through the coils of the cold storage room where it takes up the heat of the room and contents and returns to the vacuum chamber to be again partially evaporated and cooled.

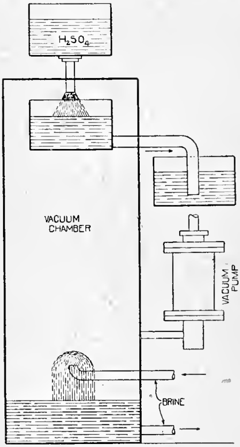


Fig. 179.

211. Cold Air System:—The cold air system is used principally on ship board. Fig. 180 shows diagrammatically the parts and the operation of the system. The cycle has four parts, *compression* in one of the cylinders of the compressor, *cooling* in the air cooler by giving off heat to the cold water

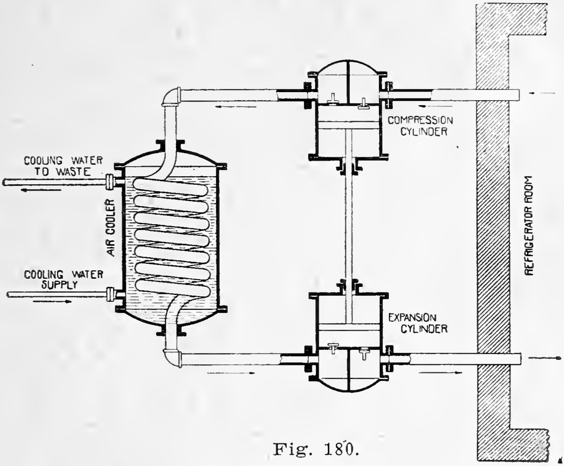


Fig. 180.

thus removing the heat of compression, *expansion* in the second cylinder of the compressor thus cooling the air, and *refrigeration* in the cold storage room where the heat lost during expansion is regained from the articles in cold-storage. Cold air machines work at low efficiencies because of the necessarily large cylinders and their attendant losses due to clearance, heating of the compression cylinder, snow in the expansion cylinder and friction. The system has much to recommend it, however, since it is extremely simple, occupies a very small space compared with other systems and uses no costly gases, chemicals or supplies.

212. The Compression and the Absorption Systems have in common this fact—both use a *refrigerant*, i. e., a liquid having a comparatively low boiling point. Perhaps the most common refrigerant is anhydrous ammonia, which boils at atmospheric pressure, at 28.5 degrees below zero and in doing so absorbs as latent heat 573 B. t. u. Table 59, Appendix, gives further properties. Other refrigerants used

to a lesser extent are sulphur dioxide, SO_2, which boils at —14 degrees under atmospheric pressure with a latent heat of 162 B. t. u. and carbon dioxide, CO_2, which boils at —30 degrees under a pressure of 182 pounds per square inch absolute with a latent heat of 140 B. t. u. A comparison of the temperatures and pressures of four common refrigerants is given in Table 64, Appendix. Pictet's fluid is a mixture of 97 per cent. sulphur dioxide and 3 per cent. carbon dioxide.

A choice of a universal refrigerant can scarcely be made because of the varying conditions of individual plants. The principal difficulty with the use of sulphur dioxide is the fact that any water uniting with it by leakage immediately produces sulphurous acid with its corroding action upon all the iron surfaces of the system. This same objection holds also for Pictet's fluid. The objections to the use of carbon dioxide are, first, its comparatively low latent heat, and second, the high pressure to which all parts of the apparatus and piping are subjected. Pressures of from 300 to 900 pounds per square inch are very common. Perhaps the worst charge that can be made against ammonia as a refrigerant is that it is highly poisonous and corrodes metals, particularly copper and copper alloys. However, the high latent heat of ammonia, together with the fact that its pressure range is neither so high as with carbon dioxide, nor so low as with sulphur dioxide, are perhaps the chief reasons for the very general preference for ammonia as the commercial refrigerant in compression systems; while its great affinity for and solubility in water, are what make the absorption system a possibility.

213. Compression System:—Compression machines may work well with the use of any one of the four refrigerants of Table 64, if the proper pressures and temperatures are observed and maintained. The common refrigerant for this type is, however, anhydrous ammonia, for reasons given above. Fig. 131 shows a diagrammatic sketch of the compression system. To follow the closed cycle of the ammonia, start with a charge being compressed in the cylinder of the compressor. From this it is conveyed by pipe to the condenser which, being cooled by water, abstracts the latent heat of the refrigerant and condenses it to a liquid. From the condenser the liquid refrigerant is conveyed to the ex-

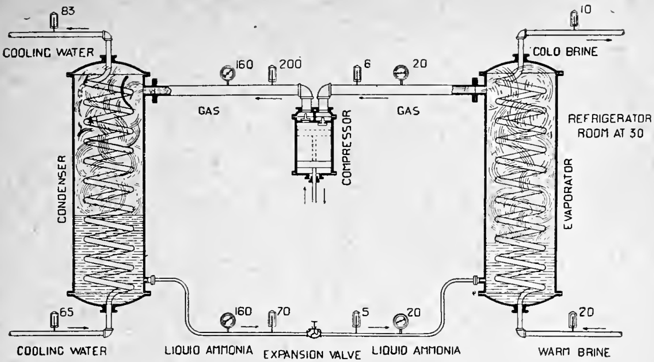


Fig. 181.

pansion valve through which it expands into the evaporator or brine cooler. In changing from a liquid to a gas in the evaporator it absorbs from the brine an amount of heat equivalent to the heat of vaporization of the ammonia. Upon leaving the evaporator the refrigerant is again ready for the cylinder of the compressor, thus completing the cycle.

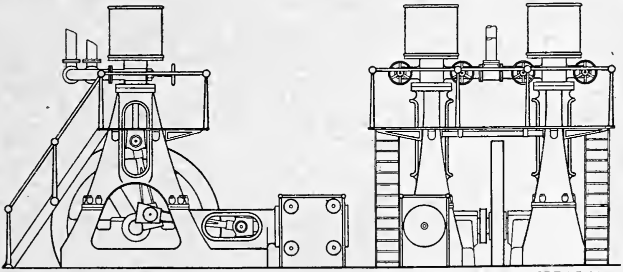


Fig. 182.

If the refrigerant is ammonia, the compressor is commonly of the vertical type, direct connected to a horizontal Corliss engine as shown in Fig. 182. This type of compressor combines the high efficiency of the Corliss engine with the vertical type of compressor which is probably the best type for reliable service of valves and pistons. The vertical compressor is usually single acting with water

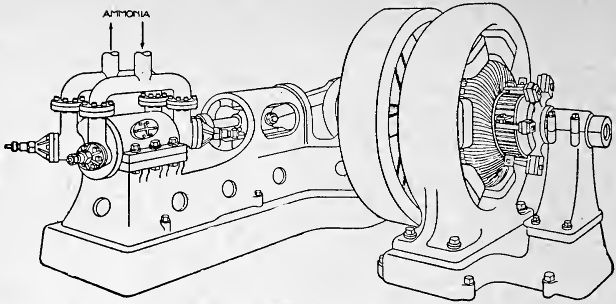


Fig. 183.

jacketed cylinders. Horizontal compressors are usually double acting, as shown in Fig. 183, where the prime mover is a direct connected electric motor. Poppet valves in this type are placed at an angle of 30 degrees to 45 degrees with the center line of the cylinder, a construction made neces-

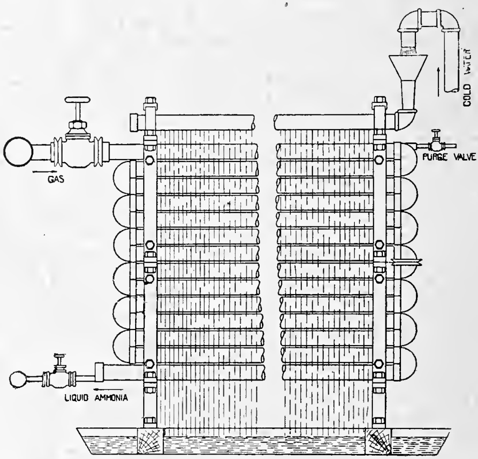


Fig. 184.

sary by space restrictions on the cylinder heads. Compressors for other refrigerants are commonly of these same types, the main difference being that compressors for carbon dioxide systems are nearly always two-stage to produce high compressions. The intermediate cooler pressures range

from 300 to 600 pounds per square inch. Horizontal steam cylinders in tandem with the compressor cylinders are common for the carbon dioxide systems and the compressor cylinders are usually single acting.

214. Condensers for Compression Systems are classified under four heads, atmospheric condensers, concentric tube condensers, enclosed condensers and submerged condensers. An elevation of an *atmospheric condenser* is shown in Fig. 184. As illustrated it consists of vertical rows of pipes so connected by return bands as to make the hot refrigerant pass through each pipe beginning at the top, while the cold water main at the top of the row furnishes a spray of water which trickles over the outside of the pipes. The gas on the inside of the pipes is thus cooled by the extraction of the quantity of heat that is used in raising the temperature of the water and evaporating a part of it. The complete con-

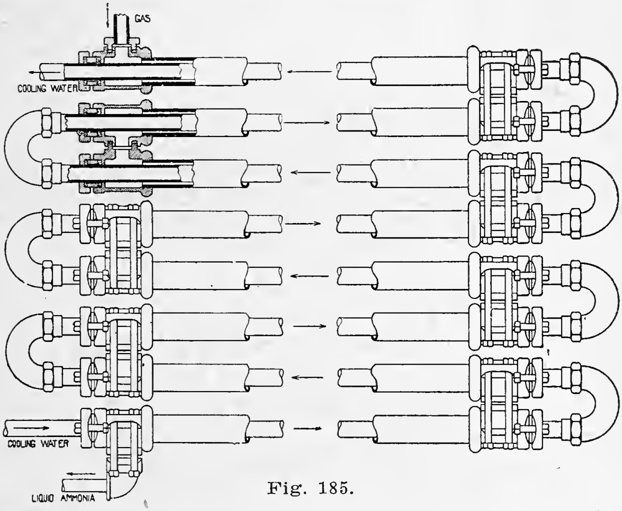


Fig. 185.

denser may consist of any required number of these vertical rows, placed side by side, each row properly connected to the hot gas header and to the liquid header.

An elevation of one section of a *concentric tube condenser* is shown in Fig. 185. The arrows show the paths of the gas and water. As in the atmospheric type the gas enters at the

top and the liquid is drawn off below. In its descent it passes through the annular space between the two concentric pipes and is cooled by the atmosphere on the outside of the larger pipes and by the water circulating through the inner pipes. This condenser has the advantage over the simple atmospheric condenser in that the water may be made to have an upward course through the apparatus, thus bringing the coldest water in contact with the pipes carrying the liquid rather than with the pipes carrying the hot gas. Since the efficiency of the plant as a whole is very largely dependent upon the temperature of the liquid at the expansion valve this matter of the "counter flow" of the cooling water is an important one. For the medium sized and large compression systems this form of condenser is used almost without exception.

The *enclosed condenser* (Fig. 186) is very similar to the surface coil condenser in steam engine plants. It consists of a cylindrical chamber with a number of concentric pipe spirals connecting a hot water header at the top with a cold water header at the bottom of the cylinder. The pipes of the spirals are provided with stuffing boxes where they pierce the upper and lower heads of the cylinder. With this condenser a counter flow of the water is used, the cold water entering the bottom of the coils and flowing upward, so that the liquid refrigerant at the bottom of the cylinder is very near the temperature of the incoming water.

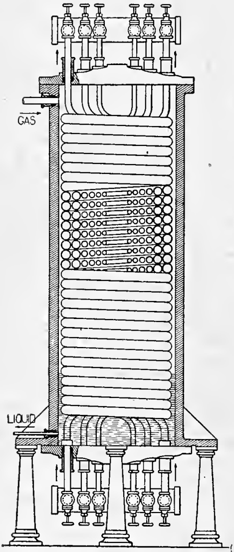


Fig. 186.

A *submerged condenser*, as the name implies, contemplates a rather large body of water below the surface of which there is submerged a coil for circulating the hot refrigerant. Fig. 187 shows a section of such a condenser. The hot gas enters at the top fitting of the coil and leaves at lower fitting. Cold water is constantly flowing in at the bot-

tom of the tank and leaving by the overflow at the top, being heated as it rises. The form of the coil is usually spiral, although this condenser may be built with coils of the return bend type when larger surface is required. Only the smaller compression plants use the enclosed or the submerged type of condenser.

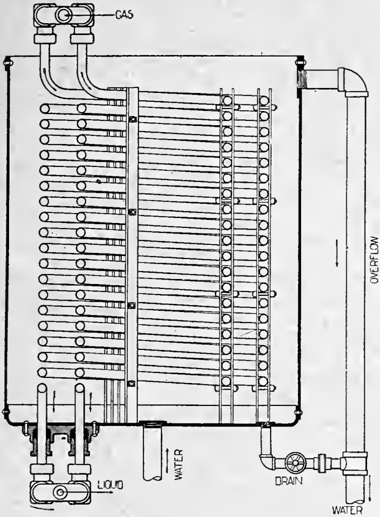


Fig. 137.

In general, condensers may be considered vital factors in the economy of compression plants. They must be reliable in service and economical in operation, and must be so designed and proportioned that they will deliver liquid refrigerant within five degrees of the temperature of the incoming cooling water. A condenser should present all joints, particularly those holding the refrigerant, to plain view for easy inspection and repair. Since it is the function of the condenser to dissipate the heat of the refrigerant gas, it is not uncommon to install it upon the roof or outside the building in some cool place. This is especially true where the atmospheric or the concentric tube types are used. In such positions the heat radiated by the condenser is not given back to the rooms and piping systems. In addition, the cooling action of the atmosphere assists in making the system more efficient.

215. Evaporators for compression systems may be considered as condensers, reversed in action but very similar in form. If the refrigerating effect is accomplished by the brine cooling system an evaporator of some type will be

necessary, but if the refrigeration is accomplished by circulating the expanding refrigerant itself, no evaporator is required. Evaporators, or brine coolers, may be classified according to the method of construction, as shell coolers and concentric tube coolers.

The *shell cooler* takes various forms. One is shown by Fig. 186, being in effect an enclosed condenser with brine instead of cold water circulating in the coils. The heat of the brine is transferred to the cool liquid refrigerant, causing the refrigerant to evaporate and take from the brine an amount of heat equal to the latent heat of the refrigerant. The proper height to which the liquid refrigerant should be allowed to rise in the evaporator is a very much disputed point, some old and experienced operators claiming greatest efficiency when about one-third of the cooling

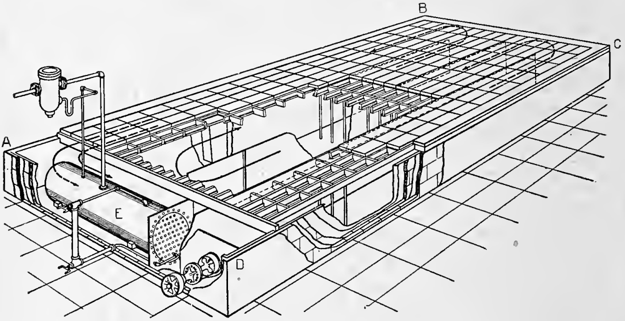


Fig. 188.

surface is covered with liquid refrigerant leaving two-thirds to be covered with gaseous refrigerant. Others claim that the entire surface should be covered or "flooded" with liquid refrigerant. These points of view give rise to the two terms *dry systems* and *flooded systems*. Of late years the flooded systems are gaining somewhat in favor, a separator being installed between the evaporator and the compressor to prevent any liquid being drawn into the compressor cylinder. This separator drains any liquid which may collect therein, back into the evaporator. In the flooded system the brine cooler more commonly takes the form shown in Fig. 188, where at the end *A D* of the brine tank *ABCD* is shown the flooded cooler *E*. This cooler consists

of a boiler shell filled with tubes, the brine circulating through the inside of the tubes while the interior of the large shell is nearly or quite filled with liquid refrigerant.

Concentric tube brine coolers are made of piping very similar in principle to that shown in Fig. 185, with the exception that instead of two concentric pipes, three are more commonly employed. The brine circulates through the innermost of the three and through the outermost, while the annular space between the smallest pipe and the middle pipe is traversed by the liquid refrigerant. In this way the annular space filled with refrigerant has brine on both sides and the cooling of the brine is very rapid. The numerous joints in this cooler present a constant source of trouble. Salt brine will usually freeze in the inner pipe, so that calcium chloride brine must be used.

A choice of evaporators or coolers depends mainly upon whether the plant is to run continuously or intermittently. When run continuously only a small amount of brine is required, and this, when cooled quickly and circulated quickly, would call for a concentric tube cooler. When run intermittently a much larger body of brine is desirable so as to remain cool longer during the night hours when the plant is not operating. For this condition a shell type cooler would probably be preferred.

In addition to the condensers and evaporators that were described in detail, there are to be found on the well equipped compression system the following pieces of apparatus which will be mentioned and described only briefly. An *oil separator* is commonly found in the line connecting the condenser with the compressor. This is simply a large cast iron cylinder with baffle plates to separate the oil from the ammonia. Since the oil is heavier than the ammonia it settles to the bottom and may be drawn off. An *ammonia scale strainer* is often found just before the compressor intake. Small *purge valves* are located at all high points in the system for the purpose of exhausting the foul gases or the air which may collect in the system. Such a purge connection is shown on the right end of the upper coil in Fig. 184.

216. Pipes, Valves and Fittings for compressor refrigerant piping are considerably different from the standard types. If the refrigerant is ammonia, no brass enters into the de-

sign of any part of the piping or auxiliaries traversed by the ammonia. The operating principles of all valves are the same as standard ones but they are made heavier and entirely of iron, or iron and aluminum. The common threaded joint used on all standard fittings is replaced in ammonia systems by the bolted and packed joint. It is not within the scope of this work to go into these details further than to give a section of an ammonia expansion valve (Fig. 189) and a section of a typical ammonia joint (Fig. 190).

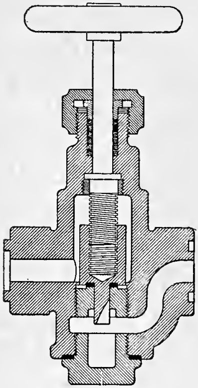


Fig. 189.

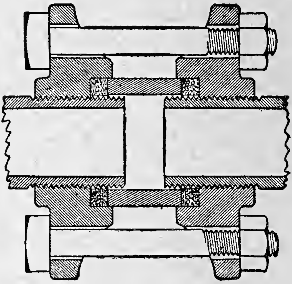


Fig. 190.

217. Absorption System:—As stated in Art. 190, the great affinity of ammonia gas for water and its solubility therein, are what make the absorption system a possibility, and give it the name as well. At atmospheric pressure and 50 degrees temperature one volume of water will absorb about 900 volumes of ammonia gas. At atmospheric pressure and 100 degrees temperature one volume of water will absorb only about one-half as much gas, or 450 volumes. If then, one volume of water is saturated at 50 degrees with ammonia gas and heated to 100 degrees there will be liberated about 450 volumes of ammonia gas. Hence it is evident that a stream of water may be used as a conveyor of ammonia gas from one place or condition to another, say from a condition of low temperature and pressure where the absorbing stream of water would be cool, to a condition of high temperature and pressure, where the gas would be liberated by simply heating the water. It will be noticed that the gas has been transferred as a liquid without a compressor or any compressive action, by pump-

ing a stream of water of approximately one-four hundred and fiftieth of the volume of the gas transferred. This, in the abstract, is the method employed in the absorption system to convey the ammonia gas from the relatively low temperature and pressure of the evaporator to the high temperature and pressure at the entrance of the condenser.

The absorption system, when closely compared in principles of operation to the compression system, differs only in one respect, namely, the absorption system replaces the gas compressor by the strong and weak liquor cycle. As shown in Fig. 191, both systems have arrangements of condenser, expansion valve and evaporator that are identical, hence the part of the cycle through these need not be considered. The problem of completing the cycle from evaporator to condenser, however, is solved quite differently in the two systems. In the compression system (upper diagram) the evaporator delivers the expanded gas to the compressor, from which, under high pressure and temperature, it is delivered to the condenser and the cycle is completed. In the absorption system (lower diagram) the evaporator delivers the expanded gas to an absorber, in which the gas comes in contact with a spray of so-called *weak liquor,* consisting of water containing about 15 to 20 per cent.

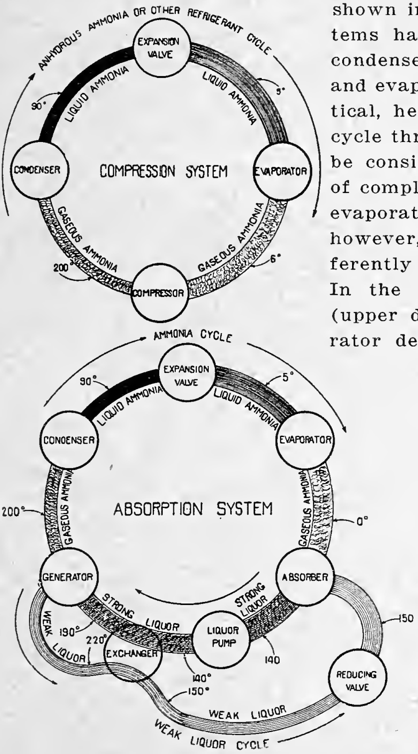


Fig. 191.

of anhydrous ammonia. The weak liquor absorbs the ammonia gas through which the liquor is sprayed and collects in the upper part of the absorber as *strong liquor*, containing about twice as much anhydrous ammonia as the weak liquor, or 30 to 35 per cent. From here it is pumped through the exchanger (which will be ignored for the present) into the generator at a pressure of about 170 pounds per square inch gage. In the generator heat is supplied by steam coils immersed in the strong liquor. As this liquor is heated it gives up about half of the contained ammonia gas which rises and passes from the generator to the condenser, thus completing the ammonia or primary cycle, while the weak liquor flows from the bottom of the generator through the exchanger and pressure reducing valve back to the absorber, thus completing the secondary or liquor cycle.

In general then, the absorption system uses two cycles, that of the ammonia and that of the liquor, the paths of the two cycles being coincident from the absorber to the generator. The liquor pump serves to keep both cycles in motion. The pump creates the pressure for both cycles and the expansion valve and the reducing valve reduce the pressure respectively for the ammonia cycle and the liquor cycle. The *exchanger* does not mix or alter the condition of the two streams of liquor passing through it, for its only function is to bring these two streams close enough that the heat of the *weak liquor from the generator* may be transferred to the *strong liquor going to the generator.* Stated in other words, the exchanger heats the strong liquor by cooling the weak liquor, thus affecting a saving of heat which would otherwise be lost, since the weak liquor must be cooled before it is ready to properly absorb the gas in the absorber.

218. An Elevation of an Absorption System with the elements piped according to what is considered best practice is shown in Fig. 192. Starting at the expansion valve, the ammonia (liquid, gas or gas in solution) passes in order through these pieces of apparatus: the evaporator, the absorber, the liquor pump, the chamber of the exchanger or the coil of the rectifier, the generator, the chamber of the rectifier and the condenser back to the expansion valve. At the same time the liquor used to absorb the gas travels in order through these pieces: the absorber, the liquor pump, the

chamber of the exchanger or the coil of the rectifier, the generator, the pressure reducing valve and the coil of the exchanger back to the absorber. The method of pipe connection shown is a very common one although some variation may be found, especially in the continued use of cooling water in consecutive pieces of apparatus. As shown, the cooling water is first used in the condenser. This will be found so in all plants. From the condenser the cooling water may next be taken to the absorber, as shown in the sketch, or it may be used in the rectifier coil instead of the strong liquor. In recent years the practice of by-passing a certain amount of the cool, strong liquor from the pump through the rectifier is gaining in favor. Fig. 192 shows a plant having bent coil construction. Plants are also built having a straight pipe construction, where all coil surfaces shown are replaced by straight pipes, the condenser being usually of the concentric tube atmospheric type and the evaporator being also of the concentric tube brine cooler type, as mentioned under compression systems. Both types of absorption plants are found in use.

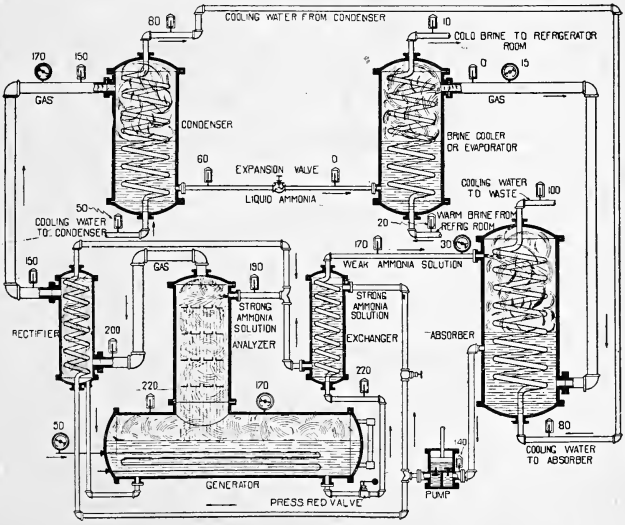


Fig. 192.

219. Generators are classified as horizontal and vertical. Fig. 193 shows a horizontal type generator, with the analyzer and exchanger, and Fig. 194 shows the vertical type, also with the analyzer. The horizontal type may have one or more horizontal cylinders equipped with steam coils. The analyzer, which may be considered as an enlarged dome of the generator, is used to condense the water vapor which rises from the surface of the liquid in the generator. To do this the analyzer has a series of horizontal baffle plates through which the incoming cool, strong liquor trickles

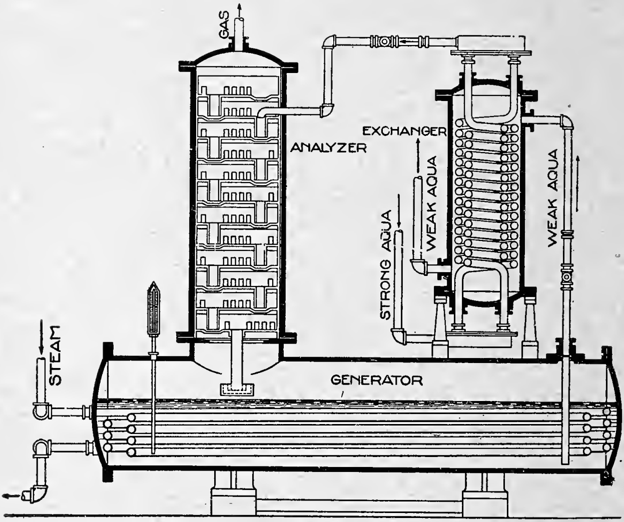


Fig. 193.

downward while the heated mixture of ammonia gas and water vapor passes upward through interstices. In this way the strong liquor gradually cools the ascending water vapor and condenses much of it on the surfaces of the baffle plates.

220. Rectifiers are arrangements of cooling surface designed to thoroughly dry the gas just before it passes into the condenser. This is accomplished by presenting to the hot product of the generator just enough cooling surface to condense the water vapor without condensing any of

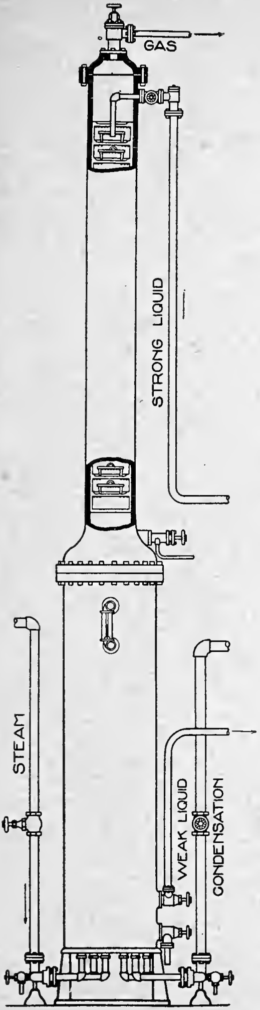


Fig. 194.

the ammonia gas. Rectifiers are very similar in general design to the various types of condensers, there being atmospheric, concentric tube, enclosed and submerged rectifiers just as there are these same type of condensers, each described under the head of condensers for compression systems. Rectifiers may save heat by the arrangement shown in Fig. 192, where the heat abstracted from the water vapor is given to the cool, strong liquor before entering the generator. As shown, the strong liquor may be divided, part passing through the rectifier and part through the exchanger, or the strong liquor may all go through the exchanger first and then through the rectifier. Where strong liquor is so used, the rectifier is always of the enclosed type. Rectifiers using water as the cooling medium are often called dehydrators, the term rectifier being more properly used when the cooling medium is the strong liquor.

221. Condensers for absorption systems do not differ in design from those used for compression systems. The same types are used, and in the same manner, the surface being somewhat less due to the precooling effect of the rectifiers or dehydrators. As a general statement, it is claimed that from 20 to 25 per cent. less surface is required in the condenser for an absorption machine than is required in one for a compression machine.

222. Absorbers may be classified as dry absorbers, wet absorbers, atmospheric absorbers, concentric tube absorbers and horizontal and vertical tubular absorbers. In the *dry absorber*, the top section of which is shown in Fig. 195, the weak liquor enters at the middle of the top header and is sprayed upon a spray pan, from which it drips downward over the coils. The gas enters as shown, part being delivered above the spray plate, so as to come into contact with the spray and the larger part being taken downward through the central pipe to a point near the bottom of the absorber, from which point it flows upward against the descending weak liquor by which it is absorbed. As the gas is dissolved by the weak liquor the heat of absorption is given off, and taken up by the cooling water in the coils. The result is a strong liquor which collects in the absorber ready to be delivered to the pump.

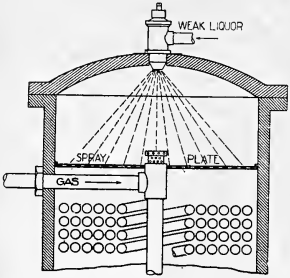


Fig.195.

The *wet absorber*, on the contrary, has practically the whole body filled with weak liquor and the ammonia gas enters near the bottom, bubbling up through the weak liquor thus saturating it. Various baffle plates with fine perforations break up the gas into small bubbles thus aiding in presenting a large surface of gas to the liquor which, as it becomes saturated and lighter, rises to the top of the body of the absorber and is ready to be drawn off by the pump. Instead of spiral cooling coils, this type is often made with straight cooling tubes inserted between two tube sheets, boiler fashion. This straight tube construction is much simpler and cheaper, and much more easily cleaned than the spiral type. It is favored by some on this account, especially where the cooling water has a tendency to form scale.

Atmospheric absorbers resemble atmospheric condensers of the single tube type. The ammonia gas and weak liquor enter the bottom through a fitting commonly called a *mixer*, and the two flow upward through the inside of the pipe while the cooling water is in contact with the outside thus taking up the heat of absorption generated within the pipes.

Concentric tube absorbers are very similar in design to concentric tube condensers, the cooling water passing through the central tube and the weak liquor and expanded gas entering at the bottom of the annular space and circulating to the top, absorption taking place on the way. Because of the small capacity of the last two mentioned absorbers, it is necessary to use with them an aqua ammonia receiver between the absorber and the ammonia pump, to act as a reservoir for storing a reserve supply of the strong liquor.

Horizontal and *vertical tubular absorbers* are those in which the cooling surface is composed of straight, horizontal or vertical tubes inserted between tube sheets, the cooling water flowing inside the tubes and the absorption taking place within the drum or body of the absorber.

223. Exchangers may be of two types, the shell type or the concentric tube type. The *shell* type, as the name implies, is composed of a main body or shell through which circulates the strong liquor to be heated and within this shell is a coil or other arrangement of heating pipes through which the hot, weak liquor flows. Fig. 192 shows the elementary arrangement of such an exchanger. Concentric tube exchangers are used on large plants. They are similar in every way to the concentric tube condensers shown in Fig. 185, with the exception that larger pipes are needed for the exchangers. The cold, strong liquor is usually carried through the pipes and the hot, weak liquor through the annular space. The great advantage of this type of exchanger is the same as that of the concentric tube condenser, namely, the counter flow of the two streams. With this arrangement the total transfer of heat is a maximum, for which reason this type of exchanger is generally preferred.

224. Coolers for the weak liquor are often found in plants. This piece of apparatus is not indicated in Fig. 192. It is usually installed as the lower three coils of the atmospheric condenser, and hence is simply a small condenser used to further cool the weak liquor just before its entrance into the absorber. With a counter flow, concentric tube exchanger a weak liquor cooler is seldom found necessary.

225. The Pump used in absorption systems to raise the pressure of the strong aqua ammonia may be steam driven, electric driven or belt driven, as best suits the particular

plant conditions. The power required by this piece of apparatus is about one horse power per 20 to 25 tons of refrigeration capacity.

226. Compression Systems and Absorption Systems Compared:—A comparison drawn between the compression system and the absorption system brings out the following facts: The compression system depends fundamentally upon the transferring of heat energy into mechanical energy and vice versa, with the attendant heavy losses. The absorption system merely transfers heat from one liquid to another. This is a process which is attended by only moderate losses. The compression system is comparatively simple, its processes readily understood and its machinery easily kept in good running order. The absorption system is complicated with a greater number of parts, its processes are often not thoroughly understood by those in charge and its machinery is likely to become inefficient because heat transferring surfaces are allowed to become dirty. For these reasons the attendance necessary upon an absorption plant must be of a higher order than that necessary for a compression plant.

227. Circulating Systems:—The refrigerating effect produced by either one of the two systems may be delivered to the place of application in two ways. The first is the *brine circulation method* wherein a brine cooler is used through which the brine flows causing the evaporation of the liquid refrigerant and the cooling of the brine. This cold brine is then circulated through pipes to the place where refrigeration is desired. Fig. 188 shows an evaporator placed in one end of a large brine tank. The refrigerating effect is carried to the cans of water by the circulation of this body of brine through the evaporator and out past the cans, the circulation through the channels shown being maintained by the pump. Brine, commonly used for such work, is made by dissolving calcium chloride in water. A 20 per cent. solution is generally used. Salt brine is used to some extent but it has many disadvantages compared with calcium brine. The second method is the *direct circulation method* wherein the liquid refrigerant is conveyed to the place to be cooled, is passed through an expansion valve and then circulated through coils in the space to be refrigerated, changing into gaseous form as fast as it can absorb enough heat. If ammonia is the refrigerant the direct circulation is not often favored because of its highly penetrative nature and odor,

even a leak so small as to escape detection being sufficient to fill the refrigerated space with the odor, which many food stuffs will absorb.

228. There are Three Methods Employed for Maintaining Low Temperatures in storage and other rooms. The first is by *direct radiation* where the pipes are placed within the room and the refrigerant is circulated through them. This is the oldest, simplest and cheapest system to install. In this the proper location and arrangement of the pipes are essential to the most efficient operation. Since the temperature to be maintained in a storage room depends upon the products to be kept in the room, it may be necessary to have a considerable range of temperature. It is desirable to have the pipes arranged as coils in two or three sets, each being valved so that the amount of refrigerant being circulated may be increased or decreased as the temperature of the stored product may require.

The pipes should be set out from the wall several inches to give free air circulation and keep the frost that collects on them from coming in contact with the wall. The coils should be so placed that the temperature of all parts of the room may be kept as nearly uniform as possible. Some products keep as well in still air as when it is in motion, but others, such as fruits, eggs, cheese, etc., are better preserved when the air is circulated. Circulation may be effected in a room piped for direct radiation by putting aprons over the coils as shown in Fig. 196. These aprons consist of 12 inch boards *D* nailed to studding *E* and the whole fastened to the coils, the studding serving to keep the boards from coming into contact with the pipe coils. A false ceiling *F* is placed a few inches below the ceiling of the room so that the warm air flows toward the pipes and over them, dropping to the floor and passing out under the lower edge of the apron into the room.

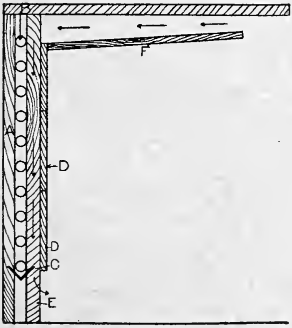


Fig. 196.

Wherever direct radiation is used drip pans should be

placed directly underneath the coils in order to catch and drain off the water when the coils are cut out and the frost melts. This water should be drained into a receptacle that can be easily emptied when filled.

The second method of room cooling is by *indirect radiation.* Let Fig. 197 represent a section of a storage building. The essential parts of the cooling system are, a bunker room *AC*, in the top part of the building, containing the cooling coils *B*, a series of ducts on either side of the building, so arranged that the air after passing over the cooling coils, drops downward. These ducts are provided with dampers for admitting as much of the cold air to the rooms as is desired. On becoming warmed this air is crowded out on the opposite side of the room into the ducts *K* and rises to the bunker-room where it is again cooled by passing over the coils. By the use of the dampers the cold air may be cut off from any room or admitted in large quantities thus making it an easy matter to maintain the temperature at any point desired. The ducts leading the air from the rooms should be 25 per cent. larger than the ones leading to the rooms and the latter should have about three square inches cross-section per square foot of floor area in rooms having a ten foot ceiling.

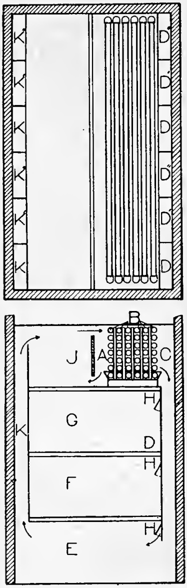


Fig. 197.

The third method is by means of a *plenum system* of air circulation, Fig. 198. The arrangements are quite similar to those of the plenum system for heating, except that the heating coils are replaced by the refrigerating coils. The air required for ventilation is blown over the coil surface, erected in a coil or bunker room, over which, oftentimes, cold water is sprayed. This not only washes the air but tends to lower its temperature. If ammonia is used as a refrigerant, brine is circulated in the coils, but if carbon dioxide is used direct expansion is employed, thus

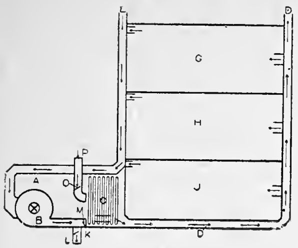


Fig. 198.

dispensing with the use of brine. The principal advantage of the plenum system of cooling is that a positive circulation of air may be maintained in any room even though the bunker room be placed on the first floor or in the basement of the building. This is the system used in large buildings that are cooled during the summer as well as heated during winter, in factories where changes of temperature seriously affect the product, as in chocolate factories, in fur storage rooms, in drying the air before it is blown into blast furnaces and in the solution of many other important economic problems.

229. Influence of the Dew Point:—In cooling a building by means of a plenum refrigerating system, great trouble is experienced with the formation of ice on the coils. For example, suppose such a cooling system on a hot summer day is taking in air at 90 degrees temperature and 85 per cent. humidity. If this air is cooled only ten degrees (see chart, page 29), it will have reached its dew point and as the cooling continues will deposit frost and ice on the coils from the liberated moisture, the air meantime remaining at the saturation point and being so delivered to the rooms. The undesirable feature of delivering saturated air to the rooms may be avoided by cooling only part, say half of the air stream, considerably lower than the final temperature desired, and then mixing it with the other half, which is drier, before delivering it to the rooms. The troublesome coating of ice and frost on the pipes may be avoided by combining the cooling system with the air washing system and using a brine spray instead of water for washing the air during cooling. The brine, which freezes at a very low temperature compared with water, plays over the cooling coils, and cleans both coils and air. The brine should preferably be a chloride brine. A modification of this method of avoiding ice and frost is to provide pans above the coils and fill them with lumps of calcium chloride. The pans have perforations so arranged that as the strong chloride

solution forms (due to the deliquescence of the salt) it trickles down over the pipes and holds the freezing point of any collecting moisture far below the temperature of the coils. A sketch of this arrangement is shown in Fig. 199, which has the disadvantage of the clumsy handling of the calcium chloride. Plants operating only during the day, as for

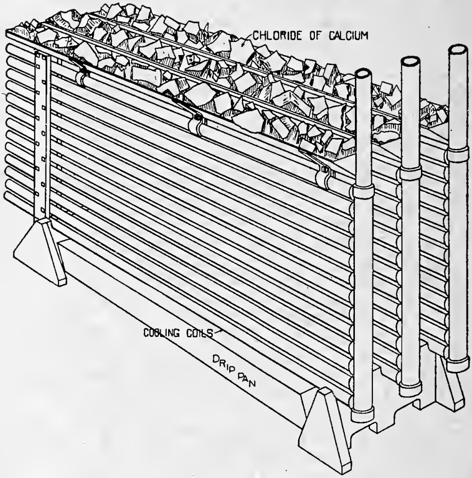


Fig. 199.

instance, auditoriums, commerce chambers, etc., often have no equipment for preventing the accumulation of frost and ice, it being allowed to form during the short period of use and to melt during the period of rest.

230. Pipe Line Refrigeration:—In a number of the larger cities refrigeration is furnished to such places as cold storage rooms, restaurants, hotels, auditoriums, etc., by a conduit system or central station system. The length of the mains in the various cities where used, ranges from a few hundred feet to twenty miles and the circulating medium employed is either liquid ammonia or brine. In the ammonia system two pipes are used, one carrying the liquid ammonia to the place desired and the other returning it after expansion to the central station. When brine is used it is good practice to circulate it at from 12 to 15 degrees F. Occasionally the conduits carry three parallel pipes, two of

which are for circulating the brine and the third is for emergency cases. The line should be divided into sections, with valves and by-passes so arranged that a defective section could be repaired without interfering with the other parts. All valves should be readily accessible and all high points in the system should be equipped with purge valves. The service pipes should be two inches in diameter and well insulated.

Either the ammonia absorption or compression system may be used for cooling the brine but according to Mr. Jos. H. Hart, the latter, making use of direct expansion, is the most efficient and the one most commonly installed. The loss by radiation to the pipes in the conduits is not great but numerous mechanical difficulties are yet to be overcome. It would seem desirable to make the pipe-line system of cooling general for residence use but as yet it has not been found economical to cool buildings using less than the equivalent of 500 pounds of refrigeration in 24 hours. Although not an efficient method, it seems probable that cold air refrigeration by using balanced expansion may supersede the other systems.

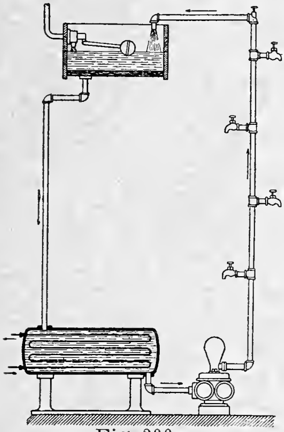


Fig. 200.

231. As a Final Application of refrigeration we may mention the cooling of the drinking water supply in large office buildings, hotels, etc. Usually this is simply a small part of the work of a large refrigerating plant. Fig. 200 gives a diagrammatic elevation of such an arrangement.

CHAPTER XVII.

REFRIGERATION CALCULATIONS

232. Unit Measurement of Refrigeration:—Since the first efforts toward refrigeration employed the simple process of melting ice by the abstraction of heat from nearby articles, it is not surprising to find the accepted standard unit for expressing refrigeration capacities referred to the refrigerating effect of a known quantity of ice. In fact, since the latent heat of fusion of ice is a constant, this furnishes an excellent basis for estimating refrigeration. The generally accepted unit of measure is the *ton of refrigeration,* which may be defined as *the amount of heat (B. t. u.) which one ton of 2000 pounds of ice at 32 degrees, will absorb in melting to water at 32 degrees.* Since the latent heat of ice is 144 B. t. u. per pound, one ton of refrigeration is equal to 288000 B. t. u. Just as a pumping plant is rated at a certain number of millions of gallons, meaning millions of gallons in twenty-four hours, so a refrigeration plant is rated in so many tons of refrigeration, meaning so many tons in twenty-four hours. Hence one ton of refrigeration capacity for one day is equivalent to 12000 B. t. u. per hour, this value being *the unit of refrigerating capacity,* sometimes referred to as *tonnage capacity,* or *refrigerating effect,* and usually designated by *T.*

233. Calculation of Required Capacity:—To estimate closely the tonnage capacity of a refrigerating plant for any certain store space requires specific attention to supplying the following losses:

(a) The radiated and conducted heat entering the room. This may be divided into that due to the walls and that due to the windows and sky-lights.

(b) The heat entering by the renewal of the air, or ventilation of the enclosed space. This may be divided into heat given off by the air and heat given off due to the latent heat of the moisture.

(c) The heat entering by the opening of doors.

(d) The heat from the men at work, lights, chemical fermentation processes, etc.

(e) The heat abstracted from material in cold storage.

Refrigeration losses due to entrance of *radiated and conducted heat* may be calculated by Equations 26, 27 and 28, Chapter III, if the proper transmission constants are inserted. To obtain these constants for various types of insulation use Tables VI and XL.

TABLE XL.

Heat Transmission of Standard Types of Dry Insulation.

Material	K	Material	K
Mill shavings, Type (a)		Hair Felt, Type (a)	
1″ thickness	.1330	1″ thickness	.138
2″ "	.1090	¼″, ½″, ¼″, Type (c)	.105
3″ "	.0920	Sheet Cork, Type (d)	
4″ "	.0800	4″ with 1″ air space	.050
5″ "	.0710	5″ with 1″ air space	.037
6″ "	.0630	3″, Type (b)	.087
7″ "	.0570	1″, Type (a)	.137
8″ "	.0520	Granulated Cork	
10″ "	.0440	4″, Type (a)	.071
12″ "	.0390	Mineral Wool	
14″ "	.0340	2½″, Type (b)	.151
16″ "	.0308	1″, Type (b)	.192
18″ "	.0279	Air Spaces	
20″ "	.0255	8″, Type (a)	.112
22″ "	.0235		
24″ "	.0218		

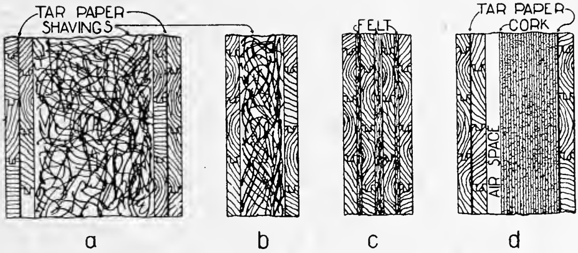


Fig. 201.

In general any space to be kept at or below zero degrees should have insulation allowing no greater transmission than .04, and for spaces to be kept at from 0 degrees to 30 degrees no greater transmission should be allowed than .06, while for temperatures above 30 degrees a transmission as great as .1 is allowable. In any case, however, it should be

remembered that the heat loss, and therefore the expense of operation, is directly proportional to this factor and the best possible insulation, consistent with available building funds, is the one to use, the ceiling and floor being as carefully insulated as the walls. Window construction should be tight, non-opening, and at least double.

The refrigeration *loss due to ventilation* may be considered under two heads, i. e., the cooling of the air from the higher to the lower temperature, and the cooling, condensing and freezing of the moisture in the air. In this particular, air cooling cannot be considered exactly the reverse of air warming. In air warming the vapor present absorbs heat but this vapor has so little heat capacity compared with that of the air that no noteworthy error is introduced by ignoring the vapor. However, in air cooling the dew point is almost invariably reached and passed, so that considerable moisture is changed from the vapor to the liquid with a liberation of its heat of vaporization. This is considerable and cannot be ignored without serious error. If, further, conditions are such that this moisture is frozen, its latent heat of freezing must also be accounted for. These two items are relatively so large that to cool air through a given range of temperature may involve several times the heat transfer required to warm the same air through the same range of temperature.

Application.—Assume outside air 95 degrees, relative humidity 85 per cent., temperature of air upon leaving cooling coils 30 degrees and temperature of coil surface 10 degrees. If 180000 cubic feet of air per hour are drawn in from the atmosphere, the refrigerating capacity of the coils may be obtained as follows. To cool the air from 95 degrees to 30 degrees will require (Equation 29),

$$\frac{180000 \times (95 - 30)}{55} = 212700 \text{ B. t. u.}$$

At 95 degrees and 85 per cent. humidity one cubic foot of air contains, (Table 11, Appendix), $.85 \times 17.124 = 14.555$ grains of moisture. At 30 degrees and saturation one cubic foot of air contains, (Table 11), 1.935 grains. Hence there would be deposited upon the coils $\frac{180000\ (14.555 - 1.935)}{7000} =$ 324.5 pounds of moisture per hour. Now there would be absorbed from each pound of this moisture

32 B. t. u. to cool from 95 to 32 degrees.
1073 B. t. u. to change to liquid form.
144 B. t. u. to freeze (if allowed to freeze on coils).
11 B. t. u. to cool from 32 to 10 degrees.

1260 B. t. u. total.

Hence the coils would have to absorb from the moisture alone $1260 \times 324.5 = 408870$ B. t. u. per hour, or for both moisture and air, $212700 + 408870 = 621600$ B. t. u. per hour. This indicates, for the ventilation proposed, a tonnage capacity of $621600 \div 12000 = 51.8$ tons of refrigeration needed at the bunker room coils. The above provides that the air is rejected at the interior temperature, 30 degrees. Modern plants, however, would pre-cool the incoming air before it reached the bunker room by having part of its heat absorbed by the outgoing 30 degree air, which would reduce the estimate somewhat below 51.8 tons.

In considering the refrigeration *loss due to the opening of doors* no rational method of calculation is applicable, but if the nature of the cold storage service is such that doors are frequently opened, as high as 25 per cent. may be allowed. Generally this is taken from 10 to 15 per cent.

The refrigeration *loss due to persons, lights,* etc., may be estimated as suggested in Art. 44. If the cooling air is recirculated, the cooling and freezing of the moisture given off by each person should be taken into account, especially if the number is large. For this purpose it is safe to assume a maximum of 500 grains of moisture given off per person per hour when such persons are not engaged in active physical exercise.

234. Calculations for Square Feet of Cooling Coil:—This problem presents greater uncertainty in its solution than does the design of a heating coil surface because of the lack of experimental data and because of the variable insulating effect of ice and frost accumulations, if allowed to form. Professor Hanz Lorenz in "Modern Refrigerating Machinery," page 349, quotes 4 B. t. u. per square foot per hour per degree difference between the average temperatures on the inside and outside of the coils, as a safe designing value when the air speed is 1000 feet per minute over the coils. This is for plants in continuous operation, as abattoirs, cold stores and in places where no provision is made against ice

formation. For clean pipe surface in the plenum air cooling plant of the New York Stock Exchange Building the heat transmission is approximately 430 B. t. u. per square foot per hour with air over coils at 1000 feet per minute. Under the average temperatures there used, this corresponds to a transmission per degree difference per square foot per hour of approximately 7 B. t. u. These two values, 4 and 7, may be taken as about the minimum and maximum transmission constants for plenum cooling coil installations.

For direct cooling coils, where the pipe surface is simply exposed to the air of the room to be cooled, Lorenz recommends a transmission allowance of not over 30 B. t. u. per square foot per hour, for in such installations the removal of ice and frost is seldom contemplated. For an average room temperature of 30 degrees and average brine temperature of 10 degrees, this would correspond to $30 \div 20 = 1.5$ B. t. u. transmitted per square foot per hour per degree difference.

APPLICATION 1.— How many lineal feet of 1¼ inch direct refrigerating coils would be required to keep a cold storage room at 30 degrees if the refrigeration loss is 80000 B. t. u. per hour total and the temperatures of the brine entering and leaving the coils are 10 degrees and 20 degrees respectively? Average brine temperature = 15 degrees. Allowing a transmission constant of 1.5, Equation 51 becomes,

$$R_r = \frac{H}{1.5\,(15 - 30)} = -\,.0445\,H$$

For this problem we have $.0445 \times 80000 = 3500$ square feet, or $3500 \times 2.3 = 8050$ lineal feet of 1¼ inch pipe.

APPLICATION 2.—The cooling of 180000 cubic feet of air per hour in Art. 233 required the extraction of 621600 B. t. u. per hour. Determine the plenum cooling surface required, if brine enters at 0 degrees and leaves at 20 degrees.

Average brine temperature = 10 degrees. Assuming that there is provision for keeping coils clear of ice, and hence a transmission constant of 7 B. t. u. is allowable, Equation 65 gives

$$R_r = \frac{621600}{7\left(10 - \dfrac{95 + 30}{2}\right)} = -\,1691 \text{ square feet of surface.}$$

The negative sign indicates a flow of heat in the direction opposite to the flow in heating installations, for which the equation was primarily designed.

235. General Application:—Considering the school building and the table of calculated results as given in Art. 155, what amount of cooling coil surface would be required to keep the temperature of all rooms of this building at 73 degrees on a day when the outside air temperature is 95 degrees and the relative humidity 85 per cent.?

Data Table XXXV gives the total heat loss of the three floors of this building as 1483250 B. t. u. per hour on a zero day when the rooms are kept at 70 degrees. Now this same building under the summer conditions would have delivered to it heat due to a temperature difference of 95 degrees — 73 degrees = 22 degrees. Hence the total refrigeration loss during the summer day would be approximately $\frac{22}{70} \times 1483250 =$ 466000 B. t. u. per hour, which amount of heat would be used to warm the incoming air from some temperature up to 73 degrees. Suppose the ventilation requirement of the building is 2000000 cubic feet per hour. Since it requires $^1/_{55}$ of a B. t. u. to warm one cubic foot of air one degree, $[2000000\ (73 - t)] \div 55 = 466000$, or $t = 60.2$, say 60 degrees. (See Arts. 50 and 51 and observe that the second term of the right hand member of Equation 36 becomes a negative term).

While the air is traversing the ducts between the coils and the rooms, allow a rise in temperature of 5 degrees. The coils would then be required to deliver 2000000 cubic feet of air per hour at 55 degrees when supplied with air at 90 degrees and 85 per cent. humidity. To cool this amount of air through the given range would require the absorption of (Equation 29) $[2000000 \times (95 - 55)] \div 55 = 1454500$ B. t. u. At 95 degrees and 85 per cent. humidity, 1 cubic foot of air contains (Table 11), $.85 \times 17.124 = 14.555$ grains of moisture. At 55 degrees and saturation point, 1 cubic foot of air contains (Table 11), 4.849 grains of moisture. Hence, neglecting change in air volume, there would be deposited on the coils approximately $[2000000\ (14.555 - 4.849)] \div 7000 = 2775$ pounds of moisture per hour.

Now, if an average brine temperature of 10 degrees is used and provision is made for keeping the coils clear of ice, the condensation will leave at some temperature above 10

degrees, say 20 degrees, and there will be absorbed from each pound of this moisture approximately

20 B. t. p. to cool from 95 to 55 degrees.
1061 B. t. p. to change to liquid form at 55 degrees.
35 B. t. u. to cool the water from 55 to 20 degrees.

1116 B. t. u. total.

Hence the coils would have to absorb from moisture alone, $2775 \times 1116 = 3096900$ B. t. u., or from both moisture and air a total of $1454500 + 3096900 = 4551400$ B. t. u. per hour. At an allowed rate of transmission of 7 B. t. u. there would be required to cool this building a total of approximately 9100 square feet of coil surface, under the conditions of ventilation as assumed.

It should be noted that whereas less than 3000 square feet of plenum surface were sufficient to heat the building according to Application 2, Art. 137, it requires fully three times this amount of surface in cooling coils to cool the building under the assumed conditions. Upon inspection it is seen that the greatest work of the cooling coils is the condensation and cooling of the moisture.

The relative humidity within the cooled rooms would be approximately 55 per cent., for the content per cubic foot of incoming air is 4.849 grains, and the capacity of the air when heated to 73 degrees is 8.782 grains showing a relative humidity, after heating, of $\frac{4.849}{8.782} = 55$ per cent. This would be raised somewhat by the persons present.

236. Ice Making Capacity. Calculation:—Neglecting losses, the ice making capacity of a refrigerating plant for a certain refrigeration tonnage may be expressed

$$I = \frac{144\ T}{(t - 32) + 144 + .5\ (32 - t_1)} \qquad (132)$$

in which I = tons of ice produced per 24 hours, T = refrigeration tonnage or rating of plant, t = initial temperature of water and t_1 = final temperature of ice, usually 12 to 18 degrees.

Application.—What should be the ice making capacity of a plant having a tonnage rating of 100, if $t = 70$ degrees and $t_1 = 16$ degrees? Take losses at 35 per cent.

$$I = \frac{.65 \times 144 \times 100}{(70 - 32) + 144 + .5\ (32 - 16)} = 49.3 \text{ tons in 24 hours.}$$

237. Gallon Degree Calculation:—For use in plants producing ice by brine circulation a unit called the *gallon degree* is sometimes used. It represents a change of one degree temperature in 1 gallon of brine in one minute of time. It is not a fixed unit representing a constant number of B. t. u., since the brine strength, and therefore its specific heat, may vary. The value in B. t. u. per minute, of a gallon degree for any plant may be obtained by multiplying the specific gravity of the brine by its specific heat and by 8.35, the weight of one gallon of water, or as an equation may be stated

$$D = 8.35\ gh \qquad (133)$$

where D = B. t. u. per minute equal to one gallon degree, g = specific gravity of brine and h = specific heat of brine.

The number of gallon degrees per ton of refrigerating capacity may be found by dividing 200 by D, since one ton of refrigerating capacity is equal to 200 B. t. u. per minute, then

$$D_t = \frac{200}{8.35\ gh} = \frac{24}{gh} \text{ for all practical purposes.} \qquad (134)$$

The refrigerating capacity of a given brine circulation may be obtained by dividing the product of the gallons circulated and the rise in brine temperature by the value D_t. Stated as an equation this is

$$T = \frac{G\ (t_2 - t_3)}{D_t} = \frac{gh\ G\ (t_2 - t_3)}{24} \qquad (135)$$

where T = tonnage capacity, G = gallons of brine circulated per minute and $(t_2 - t_3)$ = rise of brine temperature.

238. Refrigerating Capacity of Brine Cooled System:—To calculate the capacity but two things are required, the amount of brine circulated, and the rise in temperature of the brine. From these the capacity may be obtained by the equation

$$T = \frac{W\ h\ (t_2 - t_3)}{12000} \qquad (136)$$

where T = tonnage capacity, W = weight of brine circulated, in pounds, h = specific heat of brine and $(t_2 - t_3)$ = rise in temperature of brine.

239. Cost of Ice Making and Refrigeration:—The cost of ice manufacture is affected principally by the following items: price and kind of fuel, kind of water, cost of labor, regularity of operation, method of estimating costs, etc.

It is found in practice to range anywhere from $0.50 to $2.50 per ton. The items making up the cost of ice manufacture are: fuel for power, labor at the plant, water, ammonia and minor supplies, maintenance of the plant, interest and taxes, and administration. Mr. J. E. Siebel in his "Compend of Mechanical Refrigeration and Engineering" gives an itemized account of the daily operating expense of a 100-ton plant with which he was connected, the plant operating 24 hours per day.

Chief engineer	$ 5.00
Assistant engineers	6.00
Firemen	4.00
Helpers	5.00
Ice pullers	9.00
Expenses	12.00
Coal at $1.10 per ton	18.00
Delivery (wholesale) 50c per ton	50.00
Repairs, etc.	3.00
Insurance, taxes, etc.	6.00
Interest on capital	20.55
Total for 100 tons of ice	$138.55

The length of time that the ice is permitted to freeze is a factor affecting the cost of production. The following figures are given for a 10-ton plant:

	Ten tons in 12 hours	Ten tons in 24 hours
Engineer	$2.50	$5.00
Fireman	1.50	3.00
Tankmen, helpers	1.50	3.00
Coal	3.00	3.00
Repairs, supplies, etc.	1.50	1.50
Total for 10 tons	$10.00	$15.50

Mr. A. P. Criswell, in "Ice and Refrigeration," gives the following approximate costs for the production of can ice per ton with coal at $2.50 per ton and with a simple distilling system. The figures are for the plant operating at full capacity and do not include cost of administration.

Capacity of plant	Cost per ton
10 tons	$1.58
20 "	1.48
30 "	1.42
40 "	1.38
50 "	1.36
70 "	1.34
100 "	1.34
120 "	1.30

Mr. Karl Wegemann states that a certain moderate sized plant of the absorption system produced ice for a number of years at an average cost of $0.85 per ton after allowing for melting and breakage. This included all charges except for interest, insurance and administration.

The following figures taken from the books of another plant show clearly the effect of demand upon the cost of manufacture.

Month	Cost per ton
January	$3.50
February	3.70
March	2.80
April	2.17
May	1.75
June	1.19
July	1.02
August	1.02
September	1.03
October	1.26
November	2.10
December	2.94

CHAPTER XVIII.

PLANS AND SPECIFICATIONS FOR HEATING SYSTEMS.

In Planning for and Executing Engineering Contracts, the responsibilities assumed by the various interested parties should be thoroughly studied. The following outline shows the relationship between these parties and the order of the responsibility.

Owner or Purchaser {
Engineer.
Superintendent and Inspector.
General contractor, Subcontractors, Foremen and Workmen.

The engineer, the superintendent and the general contractor occupy positions of like responsibility with relation to the purchaser. The first two work for the interest of the purchaser to obtain the best possible results for the least money, and the last endeavors to fulfill the contract to the satisfaction of the superintendent, at the least possible expense to himself. These points of view are quite different and sometimes are antagonistic, but both are right and just. Of the three parties, the engineer has the greatest responsibility. It is his duty to draw up the plans and to write the specifications in such a way that every point is made clear and that no question of dispute may arise between the superintendent and the contractor. His plans should detail every part of the design with full notes. His specifications should explain all points that are difficult to delineate on the plans. They should give the purchaser's views covering all preferences, and should definitely state where and what materials may be substituted. Where any point is not definitely settled and left to the judgment of the contractor, he may be expected to interpret this point in his favor and use the cheapest material that in his judgment will give good results. This opinion may differ from that held by the purchaser. All parts should be made so plain that no two opinions could be had on any important point. The engineer

should also be careful that the plans and specifications agree in every part. The inspector is the superintendent's representative on the grounds and is supposed to inspect and pass upon all materials delivered on the grounds, and the quality of workmanship in installing. For such information see Byrne's "Inspector's Pocket-Book." The general contractor usually sublets parts of the contract to subcontractors who work through the foreman and workmen to finish the work upon the same basis as the general contractor.

The following brief set of specifications are not considered complete but are merely inserted to suggest how such work is done.

Typical Specifications.

TITLE PAGE :—

SPECIFICATIONS
for the
MATERIALS AND WORKMANSHIP
Required to Install

____

(Type of system)
HEATING AND VENTILATING SYSTEM
in the

____

(Building)

____

(Location)
by

____

(Name of designer)

INDEX PAGE :—

(To be compiled after the specifications are written.)

General Remarks to Contractor.—In the following specifications, all references to the Owner or Purchaser will mean ....or any person or persons delegated by ....to serve as the representative. The *Superintendent of Buildings* will be the purchaser's representative at all times, unless otherwise definitely stated. The contractor will, therefore, refer all doubtful questions or misunderstandings, if any, to the superintendent whose decision will be final. In case of any doubt concerning the meaning of

any part of the plans or specifications, the contractor shall obtain definite interpretation from the superintendent before proceeding with the work.

These specifications with the accompanying plans and details (sheets .... to .... inclusive) cover the purchase of all the materials as specified later (the same materials to be new in every case), and the installation of the same in a first class manner within the above named building, located at .... (street) .... (city) .... (state).

It will be understood that the successful bidder, hereinafter called the contractor, shall work in conformity with these plans and specifications and shall, to the best of his ability, carry out their true intent and meaning. He shall purchase and erect all materials and apparatus required to make the above system complete in all its parts, supplying only such quality of materials and workmanship as will harmonize with a first class system and develop satisfactory results when working under the heaviest service to which such plants are subjected.

The contractor shall lay out his own work and be responsible for its fitting to place. He shall keep a competent foreman on the grounds and shall properly protect his work at all times, making good any damage that may come to it, or to the building, or to the work of other contractors from any source whatsoever, which may be chargeable to himself or to his employees in the course of their operations.

Any defects in materials or workmanship, other than as stated under—(state exceptions if any)—that may develop within one year, shall be made good by the contractor upon written notification from the purchaser without additional cost to the purchaser.

The contractor shall, wherever it is found necessary, make all excavations and back-fill to the satisfaction of the superintendent.

The contractor shall be responsible for all cuttings of wood work, brick work or cement work, found necessary in fitting his materials to place, either within or without the building; the cutting to be done to the satisfaction of the superintendent. The contractor shall be required to connect and supply water and gas for building purposes, and shall assume all responsibility for the same.

The contractor shall be required to protect the purchaser from damage suits, originating from personal injuries received during the progress of the work; also, from actions at law because of the use of patented articles furnished by the contractor; also, from any lien or liens arising because of any materials or labor furnished.

The purchaser reserves the right to reject any or all bids.

No changes in these plans and specifications will be allowed except upon written agreement signed by both the contractor and the purchaser's representative.

System.—Specify the system of heating in a general way; high pressure, low pressure or vacuum; direct, direct-indirect or indirect radiation; basement or attic mains; one- or two-pipe connections to radiators. If ventilation is provided, state the movement of the air and the general arrangement of fans, coils or other heating surfaces. Single or double duct air lines, etc.

Boilers.—Specify type, number, size and capacity, steam pressure, approximate horse-power, heating surface, grate surface and kind of coal to be used. Locate on plan and elevation. Explain method of setting, portable or brick. Specify also, flue connection, heating and water pipe connections, kind of grate, thermometers, gages, automatic damper connection, firing tools and conditions of final tests.

Conduits and Conduit Mains.—(In this it is assumed that the boilers are not within the building). In addition to the layout, give sections of the conduit on plans showing method of construction, supporting and insulating pipes, and drainage of pipes and conduits. Specify quality and size of materials, pitch and drainage of pipes and all other points not specially provided for in the plans.

Anchors.—Locate and draw on plans and specify for the installation regarding quality of materials.

Expansion Joints or Take-ups.—Locate and draw on plans. Select type of joint and specify for amount of safe take-up and for quality of material.

Mains and Returns.—Trace the steam from the point where it enters the main, through all the special fittings of the .... system. Show where the condensation is dripped to the returns through traps or separating devices. Specify amount and direction of pitch, kind of fittings (flanged or

screwed, cast iron or malleable iron), kind of corners (long or short), method of taking up expansion and contraction. Trace returns and specify dry or wet.

Branches to Risers.—Take branches from top of mains by tees, short nipples and ells, and enter the bottom of the risers by sufficient inclination to give good drainage.

Risers.—Locate risers according to plan. They shall be straight and plumb and shall conform to the sizes given on the plans. No riser shall overlap the casing around windows. State how branches are to be taken off leading to radiators, relative to the ceiling or floor.

Radiator Connections.—Specify, one-pipe or two-pipe, number and kind of valves, sizes of connections and hand or automatic control. All connections shall allow for good drainage and expansion. Distinguish between wall radiator and floor radiator connections. If automatic control is used, hand valves at the radiators are usually omitted.

Radiators.—Specify floor or wall radiators, with type, height, number of columns and number of sections. If other radiators are substituted for the ones that are referred to as acceptable, they must be of equal amount of surface and acceptable to the superintendent. Specify brackets for wall radiators, also, air valves for all radiators, stating type and location on the radiator. Require all radiators to be cleaned with water or steam at the factory and plugged at inlet and outlet for shipment.

Piping.—Define quality, weight and material in all mains, branches and risers. All sizes above one and one-half inch are usually lap welded. Piping should be stood on end and pounded to remove all scale before going into the system. All pipes 1 inch and smaller should be reamed out full size after cutting.

Fittings.—Specify quality of fittings, whether light, standard or heavy, malleable or cast iron. Fittings with imperfect threads should be rejected.

Valves.—Specify type (globe, gate or check), whether flanged or screwed, rough or smooth body, cast iron or brass, and give pressure to be carried. All valves should be located on the plans.

Expansion Tank.—Specify capacity of tank in gallons, kind of tank (square or round, wood or steel), method of connecting up with fittings and valves, and locate definitely on plan

and elevation. Connect also to fresh water supply and to overflow.

Hangers and Ceiling Plates.—Wall radiators and horizontal runs of pipe shall be supported on suitable expansion hangers or wall supports that will permit of absolute freedom of expansion. Supports shall be placed .... feet centers. Pipe holes in concrete floors shall be thimbled. Holes through wooden walls and floors shall have suitable air space around the pipe, and all openings shall be covered with ornamental floor, ceiling or wall plates.

Traps.—Specify type, size, capacity and location. State whether flanged or screwed fittings are used and whether by-pass connection will be put in. Refer to plans.

Pressure Regulating Valve.—Specify type, size and location, also maximum and minimum steam pressure, with guarantee to operate under slight change of pressure. State if by-pass should be used and explain with plans.

Separators.—Specify type (horizontal or vertical), also size and location.

Automatic Control.—The contractor will be held responsible for the installation of all thermostats, regulator valves, air compressor, piping and fittings required to equip all rooms and halls with an automatic .... temperature control system. Specify approximate location and number of thermostats with the desired finish. Specify in a general way, regulator valves on radiators, quality of pipe, maximum test pressure for pipe, power for air supply (hydraulic, pneumatic, etc.), and supply tank. All materials in the temperature control system shall be guaranteed first class by the manufacturer through the contractor, and the system shall be guaranteed to give perfect control for a period of (two) years.

Fans.—Specify for direct connected or belt driven, right or left hand, capacity, size, housing, direction of discharge, horse-power, *R. P. M.* and pressure. State in a general way the requirements of the fan wheel, steel plate housing, shaft, bearings and the method of lubrication.

Engine.—Specify type, horse-power, steam pressure, approximate cut-off, speed and kind of control.

Electric Motors.—Specify type, horse-power, voltage, cycles, phases and *R. P. M.*

Indirect Heating Surface.—Specify the kind of surface to be put in and then state the total number of square feet of surface, with the width, height and depth of the heater. State definitely how the heaters will be assembled, giving free height of heater above the floor. Describe damper control, steam piping to and from heater, housing around heater, connection from cold air inlet to heater and connection from heater to fan. See plans. The contractor will usually follow installation instructions given by the manufacturers for the erection of the heater and engine, consequently the specifications should bear heavily only upon those points which may be varied to suit any condition. All valves, piping and fittings in this work should be controlled by the general specifications referring to these parts.

Foundations.—Specify materials and sizes.

Air Ducts, Stacks and Galvanized Iron Work.—The drawings should give the layout of all the air lines, giving connections between the air lines and the fan, and the air lines and the registers. Where these air lines are below the floor, the conduit construction should be carefully noted. All galvanized iron work should be shown on the plans and the quality and weight should be specified. Air lines should have long radius turns at the corners.

Registers.—Specify height above floor, nominal size of register, method of fitting in wall, the finish of the register and whether fitted with shutters or not.

Protection and Covering.—Specify kind and quality of pipe covering and the finish of the surface of the covering. State the amount of space between heating pipes and unprotected woodwork. Distinguish between pipes that are to be covered and those that are to be painted. All radiators and piping not covered should be painted with two coats of .... bronze or other finish acceptable to the superintendent.

Completion.—Require all rubbish removed from the building and immediate grounds and deposited at a definite place.

SUGGESTIONS TO SCHOOL DISTRICTS.

It frequently happens that School Boards and School Trustees are required to select from a number of proposed heating and ventilating systems that one which is best suited to their needs. Public officials, as a rule, are not expected to be engineers and occasionally make a selection which afterward proves to be a misfit. The following suggestions, therefore, are offered in the hope that some men may be benefited thereby.

DEFINITIONS.

Radiating surfaces.—Radiators, coils and stoves.

Circulating air.—Air which passes over the radiating surfaces and carries heat to the rooms. This may be outside air or returned air from the rooms.

Ventilating air.—Circulating air taken from without the building.

Direct system.—Radiating surfaces set within the rooms to be heated. No special outside air connections.

Indirect system.—Radiating surfaces set in the basement or somewhere within the building and below the rooms to be heated. Air is passed over these radiating surfaces, heated and carried through ducts to the rooms. When this air is taken from without the building there is provision for good ventilation.

Direct-indirect system.—Radiating surfaces set within the rooms to be heated, usually next the outside wall, and supplying ventilating air for the rooms by means of short ducts through the walls. A fair quality of ventilation may be obtained under favorable conditions.

One-pipe steam system.—Radiators connected at one bottom end only. This serves both for steam inlet and condensation return. Air valves are a necessity and are located about mid-height on the last coil of the radiator.

Two-pipe steam system.—Radiators double connected; bottom for return, and top or bottom for steam. Where top-connected for steam, use water-type radiator only. Air valves useful but not as necessary as on one-pipe system.

Low-pressure steam system.—Steam circulated by gravity at pressures between 0 and 5 pounds gage.

Atmospheric and vapor systems.—Steam circulated by gravity at 0 to 0-pressures. Radiators always two-pipe, water-type, top-connected for steam. Graduated inlet valve. Air valves on end of return.

Mechanical vacuum systems.—Steam pressures 0 to 5 pounds gage. Condensation returned by pump below atmos-

pheric pressure. Circulation positive. Returns smaller than gravity returns. Air valve on return tank.

Gravity room-heater system.—Large metal-encased stoves set within the rooms to be heated and circulating room air or outside air upward between the stove and casing, thus heating it for room use. When the air is partially or wholly taken from the outside of the building, this heater makes a direct-indirect system.

Aspirating vent flues.—Smooth vertical flues containing heating coils and leading to the outside air through the roof. These flues should be located in the partition walls of the building. The heating coils create ascending currents of air within the flues and assist room ventilation.

Cowls.—Metal cappings on the tops of the ventilating flues, so constructed as to assist convection currents within the flues and prevent down drafts.

Gravity warm-air furnace system.—A large metal- or brick-encased stove, usually set in the basement, and having one or more circulating air pipes leading into and from the space between the stove and the casing. Circulation is maintained wholly by the difference in weight (density) between the warmer air at the furnace and that of the cooler surrounding atmosphere. The entire circulating air may be returned from the rooms to the furnace, in which case there is no ventilating effect; or, a part of the air may be recirculated with part taken from the outside, giving some ventilating effect; or, all the air may be taken from the outside, with the best ventilating effect. All furnace systems should have outside air connections of such size as will permit all the air to be taken from the outside.

Fan-furnace systems.—A hot air furnace, similar in principle to the gravity warm-air furnace, with blower attachment. Circulating air temperatures generally higher than those of the warm-air furnace system.

Fan-coil system.—Metal-encased steam-coils as heating surfaces, with air circulation over the coils maintained by blower attachment. Best ventilating possibilities. Circulating air temperatures about the same as in the gravity warm-air furnace system.

Ventilating systems.—These may be either independent of or a part of the heating system. The air supply for ventilation shall be from an uncontaminated source, or shall be air from which the dust or other impurities shall be removed by efficient air cleansing devices. Circulation may be produced by gravity, in which case the air movement is slug-

gish and the ducts and stacks necessarily large; or by mechanical means, resulting in higher air velocities and correspondingly smaller ducts and stacks. Positive air circulation in vent stacks is very important in all gravity air circulating plants. Where electric power is available, mechanical circulation by fans is the most satisfactory and is the least expensive system to operate. Where power is not available, circulation may be obtained by the use of aspirating coils and cowls. Although aspiration is a wasteful process, it is practically fool-proof. In large plants where the exhaust steam may be used in coils for heating, steam engine-driven fans for the air supply are more economical than electric drives. Gas engine drives are noisy and unreliable and should be used in school systems only as a last resort.

CLASSIFICATION OF SYSTEMS.

The following classification is suggestive only, and is intended as an aid to show those systems best adapted to the different types of school buildings.

One-room building, no basement.

Direct-indirect room-heater.

One-room building, with basement.

Gravity furnace system.

Indirect, one-pipe or two-pipe steam system.

Direct-indirect, one-pipe or two-pipe steam system.

Two-room building, one floor, no basement.

Direct-indirect room heater in each room.

Two-room building, one floor, with basement.

Gravity furnace system with vent flues and cowls.

Indirect, one-pipe or two-pipe steam system with aspirating vent flues and cowls.

Direct-indirect, one-pipe or two-pipe steam system, with aspirating vent flues and cowls.

Three-room building, one floor, with basement.

Gravity furnace system with vent flues and cowls.

Indirect, one-pipe or two-pipe steam system, with aspirating vent flues and cowls.

Direct-indirect, one-pipe or two-pipe steam system, with aspirating vent flues and cowls.

Four-room building, two floors, with basement.

Gravity furnace system with vent flues and cowls.

Indirect or direct-indirect steam systems, with aspirating vent flues and cowls.

Fan furnace system (where electric power is available). Automatic temperature control.

Four-room building, two floors with basement rooms used for school purposes but not as laboratories or class rooms.

Indirect or direct-indirect steam system on first and second floors, and direct system in basement; with aspirating vent flues and cowls. With or without automatic temperature control.

Fan-furnace system (where electric power is available). Automatic temperature control.

Six- or eight-room building, with basement rooms used for laboratory and school purposes other than class rooms.

Fan-coil system, with electric power or low-pressure steam engine. Direct radiation in corridors, toilet and wash rooms. Automatic temperature control. Toilets separately ventilated by motor driven suction fans.

Direct-indirect, vapor or low-pressure steam system on first and second floors, and direct system in the basement; with aspirating vent flues and cowls. With or without automatic temperature control.

Fan-furnace system (where electric power is available). Automatic temperature control. Toilets separately ventilated by motor driven suction fans.

Moderately large buildings with basement school rooms.

Heat by direct radiation, mechanical vacuum returns; ventilate by fan-coil system. With or without air conditioning apparatus. Toilets separately ventilated by motor driven suction fans. Automatic temperature control.

Heat and ventilate by fan-coil system. With or without air conditioning apparatus. Toilets separately ventilated by motor driven suction fans. Automatic temperature control.

Large buildings with basement school rooms.

Heat by direct radiation, mechanical vacuum returns, ventilate by fan-coil system. Air conditioning apparatus. Toilets separately ventilated by motor driven suction fans. Automatic temperature control.

Heat and ventilate by fan-coil system. Air conditioning apparatus. Toilets separately ventilated by motor driven suction fans.

APPENDIX

I.

GENERAL TABLES.

HEATING AND VENTILATION.

Tables in body of text are numbered in Roman numerals, those in the Appendixes are numbered in Arabic numerals.

All tables that are not considered general are credited and added by permission of the authors.

TABLE 1.

Squares, Cubes, Sq. Roots, Cube Roots, Circles.

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circumf.	Area
.1	.010	.001	.316	.464	.314	.00785
.2	.040	.008	.447	.585	.628	.03146
.3	.090	.027	.548	.669	.942	.07069
.4	.160	.064	.633	.737	1.257	.12566
.5	.250	.125	.707	.794	1.570	.19635
.6	.360	.216	.775	.843	1.885	.28274
.7	.490	.343	.837	.888	2.200	.38485
.8	.640	.512	.894	.928	2.513	.50266
.9	.810	.729	.949	.965	2.830	.63620
1.0	1.000	1.000	1.000	1.000	3.1416	.7854
1.1	1.210	1.331	1.0488	1.0323	3.456	.9503
1.2	1.440	1.730	1.0955	1.0627	3.770	1.1310
1.3	1.690	2.197	1.1402	1.0914	4.084	1.3273
1.4	1.960	2.744	1.1832	1.1187	4.398	1.5394
1.5	2.250	3.375	1.2247	1.1447	4.712	1.7672
1.6	2.560	4.096	1.2649	1.1696	5.027	2.0106
1.7	2.890	4.913	1.3038	1.1935	5.341	2.2698
1.8	3.240	5.832	1.3416	1.2164	5.655	2.5447
1.9	3.610	6.859	1.3784	1.2386	5.969	2.8353
2.0	4.000	8.000	1.4142	1.2599	6.283	3.1416
2.1	4.410	9.261	1.4491	1.2806	6.597	3.4636
2.2	4.840	10.648	1.4832	1.3006	6.912	3.8013
2.3	5.290	12.167	1.5166	1.3200	7.226	4.1548
2.4	5.760	13.824	1.5492	1.3389	7.540	4.5239
2.5	6.250	15.625	1.5811	1.3572	7.854	4.9087
2.6	6.760	17.576	1.6125	1.3751	8.168	5.3093
2.7	7.290	19.683	1.6432	1.3925	8.482	5.7256
2.8	7.840	21.952	1.6733	1.4095	8.797	6.1575
2.9	8.410	24.389	1.7029	1.4260	9.111	6.6052
3.0	9.000	27.000	1.7321	1.4422	9.425	7.0688
3.1	9.610	29.791	1.7607	1.4581	9.739	7.5477
3.2	10.240	32.768	1.7889	1.4736	10.053	8.0425
3.3	10.890	35.937	1.8166	1.4888	10.367	8.5530
3.4	11.560	39.304	1.8439	1.5037	10.681	9.0792
3.5	12.250	42.875	1.8708	1.5183	10.996	9.6211
3.6	12.960	46.656	1.8974	1.5326	11.310	10.179
3.7	13.690	50.653	1.9235	1.5467	11.624	10.752
3.8	14.440	54.872	1.9494	1.5605	11.938	11.341
3.9	15.210	59.319	1.9748	1.5741	12.252	11.946
4.0	16.000	64.000	2.0000	1.5870	12.566	12.566
4.1	16.810	68.921	2.0249	1.6005	12.881	13.203
4.2	17.640	74.088	2.0494	1.6134	13.195	13.854
4.3	18.490	79.507	2.0736	1.6261	13.509	14.522
4.4	19.360	85.184	2.0976	1.6386	13.823	15.205

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle Circumf.	Circle Area
4.5	20.250	91.125	2.1213	1.6510	14.137	15.904
4.6	21.160	97.336	2.1448	1.6631	14.451	16.619
4.7	22.090	103.823	2.1680	1.6751	14.765	17.349
4.8	23.040	110.592	2.1909	1.6869	15.080	18.096
4.9	24.010	117.649	2.2136	1.6985	15.394	18.859
5.0	25.000	125.000	2.2361	1.7100	15.708	19.635
5.1	26.010	132.651	2.2583	1.7213	16.022	20.428
5.2	27.040	140.608	2.2804	1.7325	16.336	21.237
5.3	28.090	148.877	2.3022	1.7435	16.650	22.062
5.4	29.160	157.464	2.3238	1.7544	16.965	22.902
5.5	30.250	166.375	2.3452	1.7652	17.279	23.758
5.6	31.360	175.616	2.3664	1.7760	17.593	24.630
5.7	32.490	185.193	2.3875	1.7863	17.907	25.518
5.8	33.640	195.112	2.4083	1.7967	18.221	26.421
5.9	34.810	205.379	2.4290	1.8070	18.536	27.340
6.0	36.000	216.000	2.4495	1.8171	18.850	28.274
6.1	37.210	226.981	2.4698	1.8272	19.164	29.225
6.2	38.440	238.328	2.4900	1.8371	19.478	30.191
6.3	39.690	250.047	2.5100	1.8469	19.792	31.173
6.4	40.960	262.144	2.5298	1.8566	20.106	32.170
6.5	42.250	274.625	2.5495	1.8663	20.420	33.183
6.6	43.560	287.496	2.5691	1.8758	20.735	34.212
6.7	44.890	300.763	2.5884	1.8852	21.049	35.257
6.8	46.240	314.432	2.6077	1.8945	21.363	36.317
6.9	47.610	328.509	2.6268	1.9038	21.677	37.393
7.0	49.000	343.000	2.6458	1.9129	21.991	38.485
7.1	50.410	357.911	2.6646	1.9220	22.305	39.592
7.2	51.840	373.248	2.6833	1.9310	22.619	40.715
7.3	53.290	389.017	2.7019	1.9399	22.934	41.854
7.4	54.760	405.224	2.7203	1.9487	23.248	43.008
7.5	56.250	421.875	2.7386	1.9574	23.562	44.179
7.6	57.760	438.976	2.7568	1.9661	23.876	45.365
7.7	59.290	456.533	2.7749	1.9747	24.190	46.566
7.8	60.840	474.552	2.7929	1.9832	24.504	47.784
7.9	62.410	493.039	2.8107	1.9916	24.819	49.017
8.0	64.000	512.000	2.8284	2.0000	25.133	50.266
8.1	65.610	531.441	2.8461	2.0083	25.447	51.530
8.2	67.240	551.468	2.8636	2.0165	25.761	52.810
8.3	68.890	571.787	2.8810	2.0247	26.075	54.106
8.4	70.560	592.704	2.8983	2.0328	26.389	55.418
8.5	72.250	614.125	2.9155	2.0408	26.704	56.745
8.6	73.960	636.056	2.9326	2.0488	27.018	58.088
8.7	75.690	658.503	2.9496	2.0567	27.332	59.447
8.8	77.440	681.473	2.9665	2.0646	27.646	60.821
8.9	79.210	704.969	2.9833	2.0724	27.960	62.211

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle Circumf.	Circle Area
9.0	81.000	729.000	3.0000	2.0801	28.274	63.617
9.1	82.810	753.571	3.0166	2.0878	28.588	65.039
9.2	84.640	778.688	3.0332	2.0954	28.903	66.476
9.3	86.490	804.357	3.0496	2.1029	29.217	67.929
9.4	88.360	830.584	3.0659	2.1105	29.531	69.398
9.5	90.250	857.375	3.0822	2.1179	29.845	70.882
9.6	92.160	884.736	3.0984	2.1253	30.159	72.382
9.7	94.090	912.673	3.1145	2.1327	30.473	73.898
9.8	96.040	941.192	3.1305	2.1400	30.788	75.430
9.9	98.010	970.299	3.1464	2.1472	31.102	76.977
10	100.000	1000.000	3.1623	2.1544	31.416	78.540
11	121.000	1331.000	3.3166	2.2239	34.558	95.033
12	144.000	1728.000	3.4641	2.2894	37.699	113.097
13	169.000	2197.000	3.6056	2.3513	40.841	132.732
14	196.000	2744.000	3.7417	2.4101	43.982	153.938
15	225.000	3375.000	3.8730	2.4662	47.124	176.715
16	256.000	4096.000	4.0000	2.5198	50.265	201.062
17	289.000	4913.000	4.1231	2.5713	53.407	226.980
18	324.000	5832.000	4.2426	2.6207	56.549	254.469
19	361.000	6859.000	4.3589	2.6684	59.690	283.529
20	400.000	8000.000	4.4721	2.7144	62.832	314.159
21	441.000	9261.000	4.5826	2.7589	65.793	346.361
22	484.000	10648.000	4.6904	2.8021	69.115	380.133
23	529.000	12167.000	4.7958	2.8439	72.257	415.476
24	576.000	13824.000	4.8990	2.8845	75.398	452.389
25	625.000	15625.000	5.0000	2.9241	78.540	490.874
26	676.000	17576.000	5.0990	2.9625	81.681	530.929
27	729.000	19683.000	5.1962	3.0000	84.823	572.555
28	784.000	21952.000	5.2915	3.0366	87.965	615.752
29	841.000	24389.000	5.3852	3.0723	91.106	660.520
30	900.000	27000.000	5.4772	3.1072	94.248	706.858
31	961.000	29791.000	5.5678	3.1414	97.389	754.768
32	1024.000	32768.000	5.6569	3.1748	100.531	804.248
33	1089.000	35937.000	5.7446	3.2075	103.673	855.299
34	1156.000	39304.000	5.8310	3.2396	106.841	907.920
35	1225.000	42875.000	5.9161	3.2710	109.956	962.113
36	1296.000	46656.000	6.0000	3.3019	113.097	1017.88
37	1369.000	50653.000	6.0827	3.3322	116.239	1075.21
38	1444.000	54872.000	6.1644	3.3620	119.381	1134.11
39	1521.000	59319.000	6.2450	3.3912	122.522	1194.59
40	1600.000	64000.000	6.3246	3.4200	125.66	1256.64
41	1681.000	68921.000	6.4031	3.4482	128.81	1320.25
42	1764.000	74088.000	6.4807	3.4760	131.95	1385.44
43	1849.000	79507.000	6.5574	3.5034	135.09	1452.20
44	1936.000	85184.000	6.6333	3.5303	138.23	1520.53

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle Circumf.	Circle Area
45	2025.000	91125.000	6.7082	3.5569	141.37	1590.43
46	2116.000	97336.000	6.7823	3.5830	144.51	1661.90
47	2209.000	103823.000	6.8557	3.6088	147.65	1734.94
48	2304.000	110592.000	6.9282	3.6342	150.80	1809.56
49	2401.000	117649.000	7.0000	3.6593	153.94	1885.74
50	2500.000	125000.000	7.0711	3.6840	157.08	1963.50
51	2601.000	132651.000	7.1414	3.7084	160.22	2042.82
52	2704.000	140608.000	7.2111	3.7325	163.36	2123.72
53	2809.000	148877.000	7.2801	3.7563	166.50	2206.18
54	2916.000	157464.000	7.3485	3.7798	169.65	2290.22
55	3025.000	166375.000	7.4162	3.8030	172.79	2375.83
56	3136.000	175616.000	7.4833	3.8259	175.93	2463.01
57	3249.000	185193.000	7.5498	3.8485	179.07	2551.76
58	3364.000	195112.000	7.6158	3.8709	182.21	2642.08
59	3481.000	205379.000	7.6811	3.8930	185.35	2733.97
60	3600.000	216000.000	7.7460	3.9149	188.50	2827.43
61	3721.000	226981.000	7.8102	3.9365	191.64	2922.47
62	3844.000	238328.000	7.8740	3.9579	194.78	3019.07
63	3969.000	250047.000	7.9373	3.9791	197.92	3117.25
64	4096.000	262144.000	8.0000	4.0000	201.06	3216.99
65	4225.000	274625.000	8.0623	4.0207	204.20	3318.31
66	4356.000	287496.000	8.1240	4.0412	207.34	3421.19
67	4489.000	300763.000	8.1854	4.0615	210.49	3525.65
68	4624.000	314432.000	8.2462	4.0817	213.63	3631.68
69	4761.000	328509.000	8.3066	4.1016	216.77	3739.28
70	4900.000	343000.000	8.3666	4.1213	219.91	3848.45
71	5041.000	357911.000	8.4261	4.1408	223.05	3959.19
72	5184.000	373248.000	8.4853	4.1602	226.19	4071.50
73	5329.000	389017.000	8.5440	4.1793	229.34	4185.39
74	5476.000	405224.000	8.6023	4.1983	232.48	4300.84
75	5625.000	421875.000	8.6603	4.2172	235.62	4417.86
76	5776.000	438976.000	8.7178	4.2358	238.76	4536.46
77	5929.000	456533.000	8.7750	4.2543	241.90	4656.63
78	6084.000	474552.000	8.8318	4.2727	245.04	4778.36
79	6241.000	493039.000	8.8882	4.2908	248.19	4901.67
80	6400.000	512000.000	8.9443	4.3089	251.33	5026.55
81	6561.000	531441.000	9.0000	4.3267	254.47	5153.00
82	6724.000	551368.000	9.0554	4.3445	257.61	5281.02
83	6889.000	571787.000	9.1104	4.3621	260.75	5410.61
84	7056.000	592704.000	9.1652	4.3795	263.89	5541.77
85	7225.000	614125.000	9.2195	4.3968	267.04	5674.50
86	7396.000	636056.000	9.2736	4.4140	270.18	5808.80
87	7569.000	658503.000	9.3274	4.4310	273.32	5944.68
88	7744.000	681472.000	9.3808	4.4480	276.46	6082.12
89	7921.000	704969.000	9.4340	4.4647	279.60	6221.14

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle Circumf.	Circle Area
90	8100.000	729000.000	9.4868	4.4814	282.74	6361.73
91	8281.000	753571.000	9.5394	4.4979	285.88	6503.88
92	8464.000	778688.000	9.5917	4.5144	289.03	6647.61
93	8649.000	804357.000	9.6437	4.5307	292.17	6792.91
94	8836.000	830584.000	9.6954	4.5468	295.31	6939.78
95	9025.000	857375.000	9.7468	4.5629	298.45	7088.22
96	9216.000	884736.000	9.7980	4.5789	301.59	7238.23
97	9409.000	912673.000	9.8489	4.5947	304.73	7389.81
98	9604.000	941192.000	9.8995	4.6104	307.88	7542.96
99	9801.000	970299.000	9.9499	4.6261	311.02	7697.69
100	10000.000	1000000.000	10.0000	4.6416	314.16	7853.98
105	11025.000	1157625.000	10.2470	4.7177	329.87	8659.01
110	12100.000	1331000.000	10.4881	4.7914	345.58	9503.32
115	13225.000	1520875.000	10.7238	4.8629	361.28	10386.89
120	14400.000	1728000.000	10.9545	4.9324	376.99	11309.73
125	15625.000	1953125.000	11.1803	5.0000	392.70	12271.85
130	16900.000	2197000.000	11.4018	5.0658	408.41	13273.23
135	18225.000	2460375.000	11.6190	5.1299	424.12	14313.88
140	19600.000	2744000.000	11.8322	5.1925	439.82	15393.80
145	21025.000	3048625.000	12.0416	5.2536	455.53	16513.00
150	22500.000	3375000.000	12.2474	5.3133	471.24	17671.46
155	24025.000	3723875.000	12.4499	5.3717	486.95	18869.19
160	25600.000	4096000.000	12.6491	5.4288	502.65	20106.19
165	27225.000	4492125.000	12.8452	5.4848	518.36	21382.46
170	28900.000	4913000.000	13.0384	5.5397	534.07	22698.01
175	30625.000	5359375.000	13.2288	5.5934	549.78	24052.82
180	32400.000	5832000.000	13.4164	5.6462	565.49	25446.90
185	34225.000	6331625.000	13.6015	5.6980	581.19	26880.25
190	36100.000	6859000.000	13.7840	5.7489	596.90	28352.87
195	38025.000	7414875.000	13.9642	5.7989	612.61	29864.77
200	40000.000	8000000.000	14.1421	5.8480	628.32	31415.93
205	42025.000	8615125.000	14.3178	5.8964	644.03	33006.36
210	44100.000	9261000.000	14.4914	5.9439	659.73	34636.06
215	46225.000	9938375.000	14.6629	5.9907	675.44	36305.03
220	48400.000	10648000.000	14.8324	6.0368	691.15	38013.27
225	50625.000	11390625.000	15.0000	6.0822	706.86	39760.78
230	52900.000	12167000.000	15.1658	6.1269	722.57	41547.56
235	55225.000	12977875.000	15.3297	6.1710	738.27	43373.61
240	57600.000	13824000.000	15.4919	6.2145	753.98	45238.93
245	60025.000	14706125.000	15.6525	6.2573	769.69	47143.52
250	62500.000	15625000.000	15.8114	6.2996	785.40	49087.39
255	65025.000	16581375.000	15.9687	6.3413	801.11	51070.52
260	67600.000	17576000.000	16.1245	6.3825	816.81	53092.92
265	70225.000	18609625.000	16.2788	6.4232	832.52	55154.59
270	72900.000	19683000.000	16.4317	6.4633	848.23	57255.53

No. Diam.	Square	Cube	Sq. Root	Cube Root	Circle	
					Circumf.	Area
275	75625.000	20796875.000	16.5831	6.5030	863.94	59395.74
280	78400.000	21952000.000	16.7332	6.5421	879.65	61575.22
285	81225.000	23149125.000	16.8819	6.5808	895.35	63793.97
290	84100.000	24389000.000	17.0294	6.6191	911.06	66051.99
295	87025.000	25672375.000	17.1756	6.6569	926.77	68349.28
300	90000.000	27000000.000	17.3205	6.6943	942.48	70685.83
305	93025.000	28372625.000	17.4642	6.7313	958.19	73061.66
310	96100.000	29791000.000	17.6068	6.7679	973.89	75476.76
315	99225.000	31255875.000	17.7482	6.8041	989.60	77931.13
320	102400.000	32768000.000	17.8885	6.8399	1005.31	80424.77
325	105625.000	34328125.000	18.0278	6.8753	1021.02	82957.68
330	108900.000	35937000.000	18.1659	6.9104	1036.73	85529.86
335	112225.000	37595375.000	18.3030	6.9451	1052.43	88141.31
340	115600.000	39304000.000	18.4391	6.9795	1068.14	90792.03
345	119025.000	41063625.000	18.5742	7.0136	1083.85	93482.02
350	122500.000	42875000.000	18.7083	7.0473	1099.56	96211.28
355	126025.000	44738875.000	18.8414	7.0807	1115.27	98979.80
360	129600.000	46656000.000	18.9737	7.1138	1130.97	101787.60
365	133225.000	48627125.000	19.1050	7.1466	1146.68	104634.67
370	136900.000	50653000.000	19.2354	7.1791	1162.39	107521.01
375	140625.000	52734375.000	19.3649	7.2112	1178.10	110446.62
380	144400.000	54872000.000	19.4936	7.2432	1193.81	113411.49
385	148225.000	57066625.000	19.6214	7.2748	1209.51	116415.64
390	152100.000	59319000.000	19.7484	7.3061	1225.22	119459.06
395	156025.000	61629875.000	19.8746	7.3372	1240.93	122541.75
400	160000.000	64000000.000	20.0000	7.3681	1256.64	125663.71
405	164025.000	66430125.000	20.1246	7.3986	1272.35	128824.93
410	168100.000	68921000.000	20.2485	7.4290	1288.05	132025.43
415	172225.000	71473375.000	20.3715	7.4590	1303.76	135265.20
420	176400.000	74088000.000	20.4939	7.4889	1319.47	138544.24
425	180625.000	76765625.000	20.6155	7.5185	1335.18	141862.54
430	184900.000	79507000.000	20.7364	7.5478	1350.88	145220.12
435	189225.000	82312875.000	20.8567	7.5770	1366.59	148616.97
440	193600.000	85184000.000	20.9762	7.6059	1382.30	152053.08
445	198025.000	88121125.000	21.0950	7.6346	1398.01	155528.47
450	202500.000	91125000.000	21.2132	7.6631	1413.72	159043.13
455	207025.000	94196375.000	21.3307	7.6914	1429.42	162597.05
460	211600.000	97336000.000	21.4476	7.7194	1445.13	166190.25
465	216225.000	100544625.000	21.5639	7.7473	1460.84	169822.72
470	220900.000	103823000.000	21.6795	7.7750	1476.55	173494.45
475	225625.000	107171875.000	21.7945	7.8025	1492.26	177205.46
480	230400.000	110592000.000	21.9089	7.8297	1507.96	180955.74
485	235225.000	114084125.000	22.0227	7.8568	1523.67	184745.28
490	240100.000	117649000.000	22.1359	7.8837	1539.38	188574.10
495	245025.000	121287375.000	22.2486	7.9105	1555.09	192442.18
500	250000.000	125000000.000	22.3607	7.9370	1570.80	196349.54

TABLE 2.

Trigonometric Functions.

Angle, degrees	Sine	Tangent		Angle, degrees	Sine	Tangent	
0.0	0.00000	0.00000	90.0	47.5	0.73728	1.0913	42.5
2.5	0.04362	0.04362	87.5	50.0	0.76604	1.1917	40.0
5.0	0.08716	0.08749	85.0	52.5	0.79335	1.3032	37.5
7.5	0.13053	0.13165	82.5	55.0	0.81915	1.4281	35.0
10.0	0.17365	0.17633	80.0	57.5	0.84339	1.5697	32.5
12.5	0.21644	0.22169	77.5	60.0	0.86603	1.7321	30.0
15.0	0.25882	0.26795	75.0	62.5	0.88701	1.9210	27.5
17.5	0.30071	0.31530	72.5	65.0	0.90631	2.1445	25.0
20.0	0.34202	0.36397	70.0	67.5	0.92388	2.4142	22.5
22.5	0.38263	0.41421	67.5	70.0	0.93969	2.7474	20.0
25.0	0.42262	0.46631	65.0	72.5	0.95372	3.1716	17.5
27.5	0.46175	0.52057	62.5	75.0	0.96593	3.7321	15.0
30.0	0.50000	0.57735	60.0	77.5	0.97630	4.5107	12.5
32.5	0.53730	0.63707	57.5	80.0	0.98481	5.6713	10.0
35.0	0.57358	0.70021	55.0	82.5	0.99144	7.5958	7.5
37.5	0.60876	0.76733	52.5	85.0	0.99619	11.430	5.0
40.0	0.64279	0.83910	50.0	87.0	0.99863	19.081	3.0
42.5	0.67559	0.91633	47.5	88.5	0.99966	38.188	1.5
45.0	0.70711	1.0000	45.0	90.0	1.0000	Infinite	0.0
	Cosine	Cotangent	Angle, degrees		Cosine	Cotangent	Angle, degrees

TABLE 3.

Equivalents of Compound Units.

1 lb. per sq. in. = 27.71 in. of water at 62° F.
2.0355 in. of mercury at 32° F.
2.0416 in. of mercury at 62° F.
2.3090 ft. of water at 62° F.
1784. ft. of air at 32° F.

1 oz. per sq. in. = 0.1276 in. of mercury at 62° F.
1.732 in. of water at 62° F.

1 in. of water at 62° F. = 0.03609 lb. or .5574 oz. per s. in.
5.196 lbs. per sq. ft.
0.0736 in. of mercury at 62° F.

1 in. of water at 32° F. = 5.2021 lbs. per sq. ft.
0.036125 lb. per sq. in.

1 in. of mercury at 62° F. = 0.491 lb. or 7.86 oz. per sq. in.
1.132 ft. of water at 62° F.
13.58 in. of water at 62° F.

1 ft. of air at 32° F. = 0.0005606 lb. per sq. in.
0.015534 in. of water at 62° F.

TABLE 4.

Properties of Saturated Steam.*

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''
1	101.83	333.0	0.00300	69.8	1034.6	1104.4
2	126.15	173.5	0.00576	94.0	1021.0	1115.0
3	141.52	118.5	0.00845	109.4	1012.3	1121.6
4	153.01	90.5	0.01107	120.9	1005.7	1126.5
5	162.28	73.33	0.01364	130.1	1000.3	1130.5
6	170.06	61.89	0.01616	137.9	995.8	1133.7
7	176.85	53.56	0.01867	144.7	991.8	1136.5
8	182.86	47.27	0.02115	150.8	988.2	1139.0
9	188.27	42.36	0.02361	156.2	985.0	1141.1
10	193.22	38.38	0.02606	161.1	982.0	1143.1
11	197.75	35.10	0.02849	165.7	979.2	1144.9
12	201.96	32.36	0.03090	169.9	976.6	1146.5
13	205.87	30.03	0.03330	173.8	974.2	1148.0
14	209.55	28.02	0.03569	177.5	971.9	1149.4
14.7	212.00	26.79	0.03732	180.0	970.4	1150.4
15	213.0	26.27	0.03806	181.0	969.7	1150.7
16	216.3	24.79	0.04042	184.4	967.6	1152.0
17	219.4	23.38	0.04277	187.5	965.6	1153.1
18	222.4	22.16	0.04512	190.5	963.7	1154.2
19	225.2	21.07	0.04746	193.4	961.8	1155.2
20	228.0	20.08	0.04980	196.1	960.0	1156.2
21	230.6	19.18	0.05213	198.8	958.3	1157.1
22	233.1	18.37	0.05445	201.3	956.7	1158.0
23	235.5	17.62	0.05676	203.8	955.1	1158.8
24	237.8	16.93	0.05907	206.1	953.5	1159.6
25	240.1	16.30	0.0614	208.4	952.0	1160.4
26	242.2	15.72	0.0636	210.6	950.6	1161.2
27	244.4	15.18	0.0659	212.7	949.2	1161.9
28	246.4	14.67	0.0682	214.8	947.8	1162.6
29	248.4	14.19	0.0705	216.8	946.4	1163.2
30	250.3	13.74	0.0728	218.8	945.1	1163.9
31	252.2	13.32	0.0751	220.7	943.8	1164.5
32	254.1	12.93	0.0773	222.6	942.5	1165.1
33	255.8	12.57	0.0795	224.4	941.3	1165.7
34	257.6	12.22	0.0818	226.2	940.1	1166.3
35	259.3	11.89	0.0841	227.9	938.9	1166.8
36	261.0	11.58	0.0863	229.6	937.7	1167.3
37	262.6	11.29	0.0886	231.3	936.6	1167.8
38	264.2	11.01	0.0908	232.9	935.5	1168.4
39	265.8	10.74	0.0931	234.5	934.4	1168.9
40	267.3	10.49	0.0953	236.1	933.3	1169.4
41	268.7	10.25	0.0976	237.6	932.2	1169.8
42	270.2	10.02	0.0998	239.1	931.2	1170.3
43	271.7	9.80	0.1020	240.5	930.2	1170.7
44	273.1	9.59	0.1043	242.0	929.2	1171.2
45	274.5	9.39	0.1065	243.4	928.2	1171.6
46	275.8	9.20	0.1087	244.8	927.2	1172.0
47	277.2	9.02	0.1109	246.1	926.3	1172.4
48	278.5	8.84	0.1131	247.5	925.3	1172.8
49	279.8	8.67	0.1153	248.8	924.4	1173.2

* Marks and Davis, Handbook.

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''
50	281.0	8.51	0.1175	250.1	923.5	1173.6
51	282.3	8.35	0.1197	251.4	922.6	1174.0
52	283.5	8.20	0.1219	252.6	921.7	1174.3
53	284.7	8.05	0.1241	253.9	920.8	1174.7
54	285.9	7.91	0.1263	255.1	919.9	1175.0
55	287.1	7.78	0.1285	256.3	919.0	1175.4
56	288.2	7.65	0.1307	257.5	918.2	1175.7
57	289.4	7.52	0.1329	258.7	917.4	1176.0
58	290.5	7.40	0.1350	259.8	916.5	1176.4
59	291.6	7.28	0.1372	261.0	915.7	1176.7
60	292.7	7.17	0.1394	262.1	914.9	1177.0
61	293.8	7.06	0.1416	263.2	914.1	1177.3
62	294.9	6.95	0.1438	264.3	913.3	1177.6
63	295.9	6.85	0.1460	265.4	912.5	1177.9
64	297.0	6.75	0.1482	266.4	911.8	1178.2
65	298.0	6.65	0.1503	267.5	911.0	1178.5
66	299.0	6.56	0.1525	268.5	910.2	1178.8
67	300.0	6.47	0.1547	269.6	909.5	1179.0
68	301.0	6.38	0.1569	270.6	908.7	1179.3
69	302.0	6.29	0.1590	271.6	908.0	1179.6
70	302.9	6.20	0.1612	272.6	907.2	1179.8
71	303.9	6.12	0.1634	273.6	906.5	1180.1
72	304.8	6.04	0.1656	274.5	905.8	1180.4
73	305.8	5.96	0.1678	275.5	905.1	1180.6
74	306.7	5.89	0.1699	276.5	904.4	1180.9
75	307.6	5.81	0.1721	277.4	903.7	1181.1
76	308.5	5.74	0.1743	278.3	903.0	1181.4
77	309.4	5.67	0.1764	279.3	902.3	1181.6
78	310.3	5.60	0.1786	280.2	901.7	1181.8
79	311.2	5.54	0.1808	281.1	901.0	1182.1
80	312.0	5.47	0.1829	282.0	900.3	1182.3
81	312.9	5.41	0.1851	282.9	899.7	1182.5
82	313.8	5.34	0.1873	283.8	899.0	1182.8
83	314.6	5.28	0.1894	284.6	898.4	1183.0
84	315.4	5.22	0.1915	285.5	897.7	1183.2
85	316.3	5.16	0.1937	286.3	897.1	1183.4
86	317.1	5.10	0.1959	287.2	896.4	1183.6
87	317.9	5.05	0.1980	288.0	895.8	1183.8
88	318.7	5.00	0.2001	288.9	895.2	1184.0
89	319.5	4.94	0.2023	289.7	894.6	1184.2
90	320.3	4.89	0.2044	290.5	893.9	1184.4
91	321.1	4.84	0.2065	291.3	893.3	1184.6
92	321.8	4.79	0.2087	292.1	892.7	1184.8
93	322.6	4.74	0.2109	292.9	892.1	1185.0
94	323.4	4.69	0.2130	293.7	891.5	1185.2
95	324.1	4.65	0.2151	294.5	890.9	1185.4
96	324.9	4.60	0.2172	295.3	890.3	1185.6
97	325.6	4.56	0.2193	296.1	889.7	1185.8
98	326.4	4.51	0.2215	296.8	889.2	1186.0
99	327.1	4.47	0.2237	297.6	888.6	1186.2

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''
100	327.8	4.429	0.2258	298.3	888.0	1186.3
102	329.3	4.347	0.2300	299.8	886.9	1186.7
104	330.7	4.268	0.2343	301.3	885.8	1187.0
106	332.0	4.192	0.2386	302.7	884.7	1187.4
108	333.4	4.118	0.2429	304.1	883.6	1187.7
110	334.8	4.047	0.2472	305.5	882.5	1188.0
112	336.1	3.978	0.2514	306.9	881.4	1188.4
114	337.4	3.912	0.2556	308.3	880.4	1188.7
116	338.7	3.848	0.2599	309.6	879.3	1189.0
118	340.0	3.786	0.2641	311.0	878.3	1189.3
120	341.3	3.726	0.2683	312.3	877.2	1189.6
122	342.5	3.668	0.2726	313.6	876.2	1189.8
124	343.8	3.611	0.2769	314.9	875.2	1190.1
126	345.0	3.556	0.2812	316.2	874.2	1190.4
128	346.2	3.504	0.2854	317.4	873.3	1190.7
130	347.4	3.452	0.2897	318.6	872.3	1191.0
132	348.5	3.402	0.2939	319.9	871.3	1191.2
134	349.7	3.354	0.2981	321.1	870.4	1191.5
136	350.8	3.308	0.3023	322.3	869.4	1191.7
138	352.0	3.263	0.3065	323.4	868.5	1192.0
140	353.1	3.219	0.3107	324.6	867.6	1192.2
142	354.2	3.175	0.3150	325.8	866.7	1192.5
144	355.3	3.133	0.3192	326.9	865.8	1192.7
146	356.3	3.092	0.3234	328.0	864.9	1192.9
148	357.4	3.052	0.3276	329.1	864.0	1193.2
150	358.5	3.012	0.3320	330.2	863.2	1193.4
152	359.5	2.974	0.3362	331.4	862.3	1193.6
154	360.5	2.938	0.3404	332.4	861.4	1193.8
156	361.6	2.902	0.3446	333.5	860.6	1194.1
158	362.6	2.868	0.3488	334.6	859.7	1194.3
160	363.6	2.834	0.3529	335.6	858.8	1194.5
162	364.6	2.801	0.3570	336.7	858.0	1194.7
164	365.6	2.769	0.3612	337.7	857.2	1194.9
166	366.5	2.737	0.3654	338.7	856.4	1195.1
168	367.5	2.706	0.3696	339.7	855.5	1195.3
170	368.5	2.675	0.3738	340.7	854.7	1195.4
172	369.4	2.645	0.3780	341.7	853.9	1195.6
174	370.4	2.616	0.3822	342.7	853.1	1195.8
176	371.3	2.588	0.3864	343.7	852.3	1196.0
178	372.2	2.560	0.3906	344.7	851.5	1196.2
180	373.1	2.533	0.3948	345.6	850.8	1196.4
182	374.0	2.507	0.3989	346.6	850.0	1196.6
184	374.9	2.481	0.4031	347.6	849.2	1196.8
186	375.8	2.455	0.4073	348.5	848.4	1196.9
188	376.7	2.430	0.4115	349.4	847.7	1197.1
190	377.6	2.406	0.4157	350.4	846.9	1197.3
192	378.5	2.381	0.4199	351.3	846.1	1197.4
194	379.3	2.358	0.4241	352.2	845.4	1197.6
196	380.2	2.335	0.4283	353.1	844.7	1197.8
198	381.0	2.312	0.4325	354.0	843.9	1197.9

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''
200	381.9	2.290	0.437	354.9	843.2	1198.1
205	384.0	2.237	0.447	357.1	841.4	1198.5
210	386.0	2.187	0.457	359.2	839.6	1198.8
215	388.0	2.138	0.468	361.4	837.9	1199.2
220	389.9	2.091	0.478	363.4	836.2	1199.6
225	391.9	2.046	0.489	365.5	834.4	1199.9
230	393.8	2.004	0.499	367.5	832.8	1200.2
235	395.6	1.964	0.509	369.4	831.1	1200.6
240	397.4	1.924	0.520	371.4	829.5	1200.9
245	399.3	1.887	0.530	373.3	827.9	1201.2
250	401.1	1.850	0.541	375.2	826.3	1201.5
260	404.5	1.782	0.561	378.9	823.1	1202.1
270	407.9	1.718	0.582	382.5	820.1	1202.6
280	411.2	1.658	0.603	386.0	817.1	1203.1
290	414.4	1.602	0.624	389.4	814.2	1203.6
300	417.5	1.551	0.645	392.7	811.3	1204.1
350	431.9	1.334	0.750	408.2	797.8	1206.1
400	444.8	1.17	0.86	422.0	786.0	1208.0
450	456.5	1.04	0.96	435.0	774.0	1209.0
500	467.3	0.93	1.08	448.0	762.0	1210.0
660	486.6	0.76	1.32	469.0	741.0	1210.0

TABLE 5.

Naperian Logarithms.

e = 2.7182818 Log e = 0.4342945 = M.

1.0	0.0000	4.1	1.4110	7.2	1.9741
1.1	0.0953	4.2	1.4351	7.3	1.9879
1.2	0.1823	4.3	1.4586	7.4	2.0015
1.3	0.2624	4.4	1.4816	7.5	2.0149
1.4	0.3365	4.5	1.5041	7.6	2.0281
1.5	0.4055	4.6	1.5261	7.7	2.0412
1.6	0.4700	4.7	1.5476	7.8	2.0541
1.7	0.5306	4.8	1.5686	7.9	2.0669
1.8	0.5878	4.9	1.5892	8.0	2.0794
1.9	0.6419	5.0	1.6094	8.1	2.0919
2.0	0.6931	5.1	1.6292	8.2	2.1041
2.1	0.7419	5.2	1.6487	8.3	2.1163
2.2	0.7885	5.3	1.6677	8.4	2.1282
2.3	0.8329	5.4	1.6864	8.5	2.1401
2.4	0.8755	5.5	1.7047	8.6	2.1518
2.5	0.9163	5.6	1.7228	8.7	3.1633
2.6	0.9555	5.7	1.7405	8.8	2.1748
2.7	0.9933	5.8	1.7579	8.9	2.1861
2.8	1.0296	5.9	1.7750	9.0	2.1972
2.9	1.0647	6.0	1.7918	9.1	2.2083
3.0	1.0986	6.1	1.8083	9.2	2.2192
3.1	1.1312	6.2	1.8245	9.3	2.2300
3.2	1.1632	6.3	1.8405	9.4	2.2407
3.3	1.1939	6.4	1.8563	9.5	2.2513
3.4	1.2238	6.5	1.8718	9.6	2.2618
3.5	1.2528	6.6	1.8871	9.7	2.2721
3.6	1.2809	6.7	1.9021	9.8	2.2824
3.7	1.3083	6.8	1.9169	9.9	2.2925
3.8	1.3350	6.9	1.9315	10.0	2.3026
3.9	1.3610	7.0	1.9459		
4.0	1.3863	7.1	1.9601		

TABLE 6.

Water Conversion Factors.*

U. S. gallons	×	8.33	= pounds.
U. S. gallons	×	0.13368	= cubic feet.
U. S. gallons	×	231.00000	= cubic inches.
U. S. gallons	×	3.78	= liters.
Cubic inches of water (39.1°)	×	0.036024	= pounds.
Cubic inches of water (39.1°)	×	0.004329	= U. S. gallons.
Cubic inches of water (39.1°)	×	0.576384	= ounces.
Cubic feet of water (39.1°)	×	62.425	= pounds.
Cubic feet of water (39.1°)	×	7.48	= U. S. gallons.
Cubic feet of water (39.1°)	×	0.028	= tons.
Pounds of water	×	27.72	= cubic inches.
Pounds of water	×	0.01602	= cubic feet.
Pounds of water	×	0.12	= U. S. gallons.

* American Machinist Hand Book.

TABLE 7.

Volume and Weight of Dry Air at Different Temperatures.*

Under a constant atmospheric pressure of 29.92 inches of mercury, the volume at 32° F. being 1.

Temp. deg. F.	Volume	Weight per cu. ft.	Temp. deg. F.	Volume	Weight per cu. ft.
0	.935	.0864	500	1.954	.0413
12	.960	.0842	552	2.056	.0385
22	.980	.0824	600	2.150	.0376
32	1.000	.0807	650	2.260	.0357
42	1.020	.0791	700	2.362	.0338
52	1.041	.0776	750	2.465	.0328
62	1.061	.0761	800	2.566	.0315
72	1.082	.0747	850	2.668	.0303
82	1.102	.0733	900	2.770	.0292
92	1.122	.0720	950	2.871	.0281
102	1.143	.0707	1000	2.974	.0268
112	1.163	.0694	1100	3.177	.0254
122	1.184	.0682	1200	3.381	.0239
132	1.204	.0671	1300	3.584	.0225
142	1.224	.0659	1400	3.788	.0213
152	1.245	.0649	1500	3.993	.0202
162	1.265	.0638	1600	4.196	.0192
172	1.285	.0628	1700	4.402	.0183
182	1.306	.0618	1800	4.605	.0175
192	1.326	.0609	1900	4.808	.0168
202	1.347	.0600	2000	5.012	.0161
212	1.367	.0591	2100	5.217	.0155
230	1.404	.0575	2200	5.420	.0149
250	1.444	.0559	2300	5.625	.0142
275	1.495	.0540	2400	5.827	.0138
300	1.546	.0522	2500	6.032	.0133
325	1.597	.0506	2600	6.236	.0130
350	1.648	.0490	2700	6.440	.0125
375	1.689	.0477	2800	6.644	.0121
400	1.750	.0461	2900	6.847	.0118
450	1.852	.0436	3000	7.051	.0114

TABLE 8.

Temperature of the Boiling Point at Different Heights of the Mercury Column.†

Inches	29.92	28.75	27.62	26.52	25.46	24.44	23.45
Temp. F.	212	210	208	206	204	202	200
Inches	22.50	21.58	20.68	19.83	19.00	18,20	17.42
Temp. F.	198	196	194	192	190	188	186

* Suplee's M. E. Reference Book.

† Smithsonian Tables.

TABLE 9.

Weight of Pure Water per Cubic Foot at Various Temperatures.*

Temp. deg. F.	Weight lbs. per cu. ft.	B. t. u. per pound above 32	Temp. deg. F.	Weight lbs. per cu. ft.	B. t. u. per pound above 32
32	62.42	0.00	77	62.26	45.04
33	62.42	1.01	78	62.25	46.04
34	62.42	2.02	79	62.24	47.04
35	62.42	3.02	80	62.23	48.03
36	62.42	4.03	81	62.22	49.03
37	62.42	5.04	82	62.21	50.03
38	62.42	6.04	83	62.20	51.02
39	62.42	7.05	84	62.19	52.02
40	62.42	8.05	85	62.18	53.02
41	62.42	9.05	86	62.17	54.01
42	62.42	10.06	87	62.16	55.01
43	62.42	11.06	88	62.15	56.01
44	62.42	12.06	89	62.14	57.00
45	62.42	13.07	90	62.13	58.00
46	62.42	14.07	91	62.12	59.00
47	62.42	15.07	92	62.11	60.00
48	62.41	16.07	93	62.10	60.99
49	62.41	17.08	94	62.09	61.99
50	62.41	18.08	95	62.08	62.99
51	62.41	19.08	96	62.07	63.98
52	62.40	20.08	97	62.06	64.98
53	62.40	21.08	98	62.05	65.98
54	62.40	22.08	99	62.03	66.97
55	62.39	23.08	100	62.02	67.97
56	62.39	24.08	101	62.01	68.97
57	62.39	25.08	102	62.00	69.96
58	62.38	26.08	103	61.99	70.96
59	62.38	27.08	104	61.97	71.96
60	62.37	28.08	105	61.96	72.95
61	62.37	29.08	106	61.95	73.95
62	62.36	30.08	107	61.93	74.95
63	62.36	31.07	108	61.92	75.95
64	62.35	32.07	109	61.91	76.94
65	62.34	33.07	110	61.89	77.94
66	62.34	34.07	111	61.88	78.94
67	62.33	35.07	112	61.86	79.93
68	62.33	36.07	113	61.85	80.93
69	62.32	37.06	114	61.83	81.93
70	62.31	38.06	115	61.82	82.92
71	62.31	39.06	116	61.80	83.92
72	62.30	40.05	117	61.78	84.92
73	62.29	41.05	118	61.77	85.92
74	62.28	42.05	119	61.75	86.91
75	62.28	43.05	120	61.74	87.91
76	62.27	44.04	121	61.72	88.91

* Kent's M. E. Pocket-Book. 8th Edition.

Temp. deg. F.	Weight lbs. per cu. ft.	B. t. u. per pound above 32	Temp. deg. F.	Weight lbs. per cu. ft.	B. t. u. per pound above 32
122	61.70	89.91	167	60.83	134.86
123	61.68	90.90	168	60.81	135.86
124	61.67	91.90	169	60.79	136.86
125	61.65	92.90	170	60.77	137.87
126	61.63	93.90	171	60.75	138.87
127	61.61	94.89	172	60.73	139.87
128	61.60	95.89	173	60.70	140.87
129	61.58	96.89	174	60.68	141.87
130	61.56	97.89	175	60.66	142.87
131	61.54	98.89	176	60.64	143.87
132	61.52	99.88	177	60.62	144.88
133	61.51	100.88	178	60.59	145.88
134	61.49	101.88	179	60.57	146.88
135	61.47	102.88	180	60.55	147.88
136	61.45	103.88	181	60.53	148.88
137	61.43	104.87	182	60.50	149.89
138	61.41	105.87	183	60.48	150.89
139	61.39	106.87	184	60.46	151.89
140	61.37	107.87	185	60.44	152.89
141	61.36	108.87	186	60.41	153.89
142	61.34	109.87	187	60.39	154.90
143	61.32	110.87	188	60.37	155.90
144	61.30	111.87	189	60.34	156.90
145	61.28	112.86	190	60.32	157.91
146	61.26	113.86	191	60.29	158.91
147	61.24	114.86	192	60.27	159.91
148	61.22	115.86	193	60.25	160.91
149	61.20	116.86	194	60.22	161.92
150	61.18	117.86	195	60.20	162.92
151	61.16	118.86	196	60.17	163.92
152	61.14	119.86	197	60.15	164.93
153	61.12	120.86	198	60.12	165.93
154	61.10	121.86	199	60.10	166.94
155	61.08	122.86	200	60.07	167.94
156	61.06	123.86	201	60.05	168.94
157	61.04	124.86	202	60.02	169.95
158	61.02	125.86	203	60.00	170.95
159	61.00	126.86	204	59.97	171.96
160	60.98	127.86	205	59.95	172.96
161	60.96	128.86	206	59.92	173.97
162	60.94	129.86	207	59.89	174.97
163	60.92	130.86	208	59.87	175.98
164	60.90	131.86	209	59.84	176.98
165	60.87	132.86	210	59.82	177.99
166	60.85	133.86	211	59.79	178.99
			212	59.76	180.

TABLE 10.

Boiling Point of Water at Different Heights of Vacuum.

Temp. F.	Height of mercury in vacuum tube in inches	Temp. F.	Height of mercury in vacuum tube in inches
212.0	0.00	175.8	16.00
210.3	1.00	172.6	17.00
208.5	2.00	169.0	18.00
206.8	3.00	165.3	19.00
204.8	4.00	161.2	20.00
202.9	5.00	156.7	21.00
200.9	6.00	151.9	22.00
199.0	7.00	146.5	23.00
196.7	8.00	140.3	24.00
194.5	9.00	133.3	25.00
192.2	10.00	124.9	26.00
189.7	11.00	114.4	27.00
187.3	12.00	108.4	28.00
184.6	13.00	102.0	29.00
181.3	14.00	98.0	29.92
178.9	15.00		

TABLE 11.

Weight of Water with Air per Cubic Foot at Different Temperatures and at Saturation.

Temp. F.	Weight, grains	Temp. F.	Weight, grains	Temp. F.	Weight, grains	Temp. F.	Weight, grains	Temp. F.	Weight, grains	Temp. F.	Weight, grains
—20	0.166	2	0.529	24	1.483	46	3.539	68	7.480	90	14.790
—19	0.174	3	0.554	25	1.551	47	3.667	69	7.726	91	15.234
—18	0.184	4	0.582	26	1.623	48	3.800	70	7.980	92	15.689
—17	0.196	5	0.610	27	1.697	49	3.936	71	8.240	93	16.155
—16	0.207	6	0.639	28	1.773	50	4.076	72	8.508	94	16.634
—15	0.218	7	0.671	29	1.853	51	4.222	73	8.782	95	17.124
—14	0.231	8	0.704	30	1.935	52	4.372	74	9.066	96	17.626
—13	0.243	9	0.739	31	2.022	53	4.526	75	9.356	97	18.142
—12	0.257	10	0.776	32	2.113	54	4.685	76	9.655	98	18.671
—11	0.270	11	0.816	33	2.194	55	4.849	77	9.962	99	19.212
—10	0.285	12	0.856	34	2.279	56	5.016	78	10.277	100	19.766
— 9	0.300	13	0.898	35	2.366	57	5.191	79	10.601	101	20.335
— 8	0.316	14	0.941	36	2.457	58	5.370	80	10.934	102	21.017
— 7	0.332	15	0.986	37	2.550	59	5.555	81	11.275	103	21.514
— 6	0.350	16	1.032	38	2.646	60	5.745	82	11.626	104	22.125
— 5	0.370	17	1.080	39	2.746	61	5.941	83	11.987	105	22.750
— 4	0.389	18	1.128	40	2.849	62	6.142	84	12.356	106	23.392
— 3	0.411	19	1.181	41	2.955	63	6.349	85	12.736	107	24.048
— 2	0.434	20	1.235	42	3.064	64	6.563	86	13.127	108	24.720
— 1	0.457	21	1.294	43	3.177	65	6.782	87	13.526	109	25.408
0	0.481	22	1.355	44	3.294	66	7.009	88	13.937	110	26.112
1	0.505	23	1.418	45	3.414	67	7.241	89	14.359		

TABLE 12.

Relative Humidities.

Difference between the dry and wet thermometers.

Air temperatures	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	Air temperatures
30	100	89	78	67	57	47	36	26	17	7																												30
35	100	91	82	73	65	54	45	37	28	19	12	3																										35
40	100	92	84	76	68	60	53	45	38	30	22	16	8	1																								40
45	100	92	85	78	71	64	58	51	44	38	32	25	19	13	7	1																						45
50	100	93	87	80	74	67	61	55	50	44	38	33	27	22	16	11	6	1																				50
55	100	94	88	82	76	70	65	59	54	49	43	39	34	29	24	19	16	10	6	1																		55
60	100	94	89	84	78	73	68	63	58	53	48	44	39	34	30	26	22	18	14	10	6	2																60
65	100	95	90	85	80	75	70	65	61	56	52	48	44	39	35	31	28	24	20	17	13	10	6	3														65
70	100	95	90	86	81	77	72	68	64	60	55	52	48	44	40	36	33	29	26	23	19	16	13	10	7	4	1											70
75	100	95	91	87	82	78	74	70	66	62	58	55	51	47	44	40	37	34	31	27	24	21	19	16	13	10	7	5	2									75
80	100	96	92	87	83	79	75	72	68	64	61	57	54	51	47	44	41	38	35	32	29	26	23	20	18	15	13	10	8	6	3	1						80
85	100	96	92	88	84	80	77	73	70	66	63	60	56	53	50	47	44	41	38	36	33	30	28	25	22	20	17	15	13	11	9	6	4	2				85
90	100	96	92	88	85	81	78	75	71	68	65	62	59	56	53	50	47	44	41	39	36	34	32	29	26	24	22	20	17	15	13	11	9	7	5	3	2	90
95	100	96	93	89	86	82	79	76	72	69	66	63	60	58	55	52	49	47	44	42	39	37	35	32	30	28	25	23	21	19	17	15	13	11	10	8	6	95
100	100	97	93	90	86	83	80	77	74	71	68	65	62	59	57	54	51	49	47	44	42	39	37	35	33	31	29	27	25	23	21	19	17	15	14	12	10	100
105	100	97	93	90	87	84	81	78	75	72	69	66	64	61	58	56	53	51	49	46	44	42	40	38	35	33	31	30	28	26	24	22	20	19	17	15	14	105
110	100	97	94	90	87	84	81	78	76	73	70	67	65	62	60	57	55	53	50	48	46	44	42	40	38	36	34	32	30	28	27	25	23	22	20	19	17	110
115	100	97	94	91	88	85	82	79	76	74	71	69	66	64	61	59	57	54	52	50	48	46	44	42	40	38	36	34	33	31	29	28	26	24	23	21	20	115
120	100	97	94	91	88	85	83	80	77	75	72	70	67	65	62	60	58	56	54	51	49	47	45	44	42	40	38	36	35	33	31	30	28	27	25	24	22	120
125	100	97	94	91	88	86	83	80	78	75	73	70	68	66	64	62	59	57	55	53	51	49	47	45	43	42	40	38	37	35	33	32	30	29	27	26	24	125
130	100	97	94	91	89	86	83	81	78	76	74	71	69	67	65	62	60	58	56	54	52	50	49	47	45	43	42	40	38	37	35	33	32	31	29	28	27	130
135	100	97	94	92	89	86	84	81	79	77	74	72	70	68	65	63	61	59	57	55	53	51	50	48	46	45	43	41	40	38	37	35	34	32	31	30	28	135
140	100	97	95	92	89	87	84	82	79	77	75	73	71	68	66	64	62	60	58	56	55	53	51	49	48	46	44	43	41	40	38	37	35	34	33	31	30	140
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	

TABLE 13.

Properties of Air with Moisture under Pressure of One Atmosphere.*

Temperature Fahrenheit	Vol. of dry air at different temp. the vol. at 32° being 1.000	Weight of a cu. ft. of dry air at different temps. in lbs.	Elastic force of vapor in inches of mercury (Regnault)	Mixtures of air saturated with vapor						Cubic feet of vapor from one pound of water at pressure as in column 4	Cubic feet dry air warmed one degree per B. t. u.
				Elastic force of the air in the mixture of air and vapor in inches of mer.	Weight of cubic foot of the mixture.			Ratio of water to dry air	Ratio of dry air to water vapor		
					Weight of the air in pounds	Weight of the vapor in pounds	Total weight of mixture in lbs.				
1	2	3	4	5	6	7	8	9	10	11	12
0	.935	.0864	0.044	29.877	.0863	.000079	.086379	.00092	1092.40	------	48.5
12	.960	.0842	0.074	29.849	.0840	.000130	.084130	.00115	646.10	------	50.1
22	.980	.0824	0.118	29.803	.0821	.000202	.082302	.00245	406.40	------	51.1
32	1.000	.0807	0.181	29.740	.0802	.000304	.080504	.00379	263.81	3289.0	52.0
42	1.020	.0791	0.267	29.654	.0784	.000440	.078840	.00561	178.18	2252.0	53.2
52	1.041	.0766	0.388	29.533	.0766	.000627	.077227	.00819	122.17	1595.0	54.0
60	1.057	.0764	0.522	29.399	.0751	.000830	.075252	.01251	92.27	1227.0	55.0
62	1.061	.0761	0.556	29.365	.0747	.000881	.075581	.01179	84.79	1135.0	55.2
70	1.078	.0750	0.754	29.182	.0731	.001153	.073509	.01780	64.59	882.0	56.2
72	1.082	.0747	0.785	29.136	.0727	.001221	.073921	.01680	59.54	819.0	56.3
82	1.102	.0733	1.092	28.829	.0706	.001667	.072267	.02361	42.35	600.0	57.2
92	1.122	.0720	1.501	28.420	.0684	.002250	.070717	.03289	30.40	444.0	58.4
100	1.139	.0710	1.929	27.992	.0664	.002848	.069261	.04495	23.66	356.0	59.1
102	1.143	.0707	2.036	27.885	.0659	.002997	.068897	.04547	21.98	334.0	59.5
112	1.163	.0694	2.731	27.190	.0631	.003946	.067042	.06253	15.99	253.0	60.6
122	1.184	.0682	3.621	26.300	.0599	.005142	.065046	.08584	11.65	194.0	61.7
132	1.204	.0671	4.752	25.169	.0564	.006639	.063039	.11771	8.49	151.0	62.5
142	1.224	.0660	6.165	23.756	.0524	.008473	.060873	.16170	6.18	118.0	63.7
152	1.245	.0649	7.930	21.991	.0477	.010716	.058416	.22465	4.45	93.3	64.7
162	1.265	.0638	10.099	19.822	.0423	.013415	.055715	.31713	3.15	74.5	65.8
172	1.285	.0628	12.758	17.163	.0360	.016682	.052682	.46338	2.16	59.2	66.9
182	1.306	.0618	15.960	13.961	.0288	.020536	.049336	.71300	1.402	48.6	68.0
192	1.326	.0609	19.828	10.093	.0205	.025142	.045642	1.22643	.815	39.8	69.0
202	1.347	.0600	24.450	5.471	.0109	.030545	.041445	2.80230	.357	32.7	70.0
212	1.367	.0591	29.921	0.000	.0000	.036820	.036820	Infinite	.000	27.1	71.1

* Carpenter's H. & V. B. and Sturtevant's Mech. Draft.

TABLE 14.

Dew-Points of Air According to Its Hygrometric State.*

Temp.		Relative moisture									
		90%		80%		70%		60%		50%	
C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.
0	32.0	— 1.5	29.3	— 3.0	26.6	— 4.9	23.2	— 6.5	20.3	— 9.2	15.4
2	35.6	0.9	33.6	— 0.9	30.4	— 2.5	27.5	— 4.8	23.4	— 7.1	19.2
4	39.2	2.4	36.3	0.9	33.6	— 0.9	30.4	— 2.9	26.8	— 5.3	22.5
6	42.8	4.5	40.1	2.9	37.2	0.9	33.6	— 1.3	29.7	— 3.7	25.3
8	46.4	6.4	43.5	4.5	40.1	2.7	36.9	0.6	33.1	— 1.9	28.6
10	50.0	8.5	47.3	6.8	44.2	4.5	40.1	2.5	36.5	0.0	32.0
12	53.6	10.5	50.9	8.5	47.3	6.8	44.2	4.3	39.7	2.0	35.6
14	57.2	12.3	54.1	10.5	50.9	8.5	47.3	6.2	43.2	3.7	38.7
16	60.8	14.4	57.9	12.6	54.7	10.5	50.9	8.3	46.9	5.6	42.1
18	64.4	16.5	61.7	14.6	58.3	12.4	54.3	10.0	50.0	7.4	45.3
20	68.0	18.3	64.9	16.5	61.7	14.4	57.9	11.9	53.4	9.2	48.6
22	71.6	20.3	68.5	18.4	65.1	16.3	61.3	13.7	56.7	11.6	52.8
24	75.2	22.2	72.1	20.5	68.9	18.4	65.1	15.6	60.0	13.0	55.4
26	78.8	24.4	75.9	22.2	72.1	20.1	68.2	17.6	63.6	14.7	58.5
28	82.4	26.3	79.3	24.2	75.6	22.0	71.6	19.5	67.1	17.5	63.5
30	86.0	28.3	82.9	26.3	79.3	23.9	75.0	21.5	70.7	18.3	64.9

* Bulletin 21, Int. Ass'n of Refrig.

Psychrometric Charts Recent Tests.

In recent years a highly technical study of humidity and its control has been made by Mr. Willis H. Carrier. Fig. A shows, merely for the sake of comparison, how closely his results checked the earlier values obtained by the Government Weather Bureau. The following charts, Figs. B and C, summarize the results of Mr. Carrier's experiments Fig. C is a part of Fig. B drawn to a larger scale.

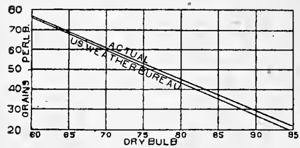


Fig. A.

As one illustration of the use of the chart, refer to Fig. C with air at 40 degrees and 40 per cent. humidity. If this air be heated to 100 degrees without addition of moisture it will be seen by interpolation that the humidity drops to about 8 per cent. If the same be heated to 100 degrees and enough moisture be added to keep the relative humidity at 40 per cent., then the absolute humidity changes from 15 grains to 120 grains per pound of air. These figures may be reduced to grains per cubic foot by dividing by the volume per pound as given in the second column and will be found to check closely with those given by Fig. 7 and Table 9. Almost any other points relating to changes in volume, humidity and contained heat may be easily worked out by these curves.

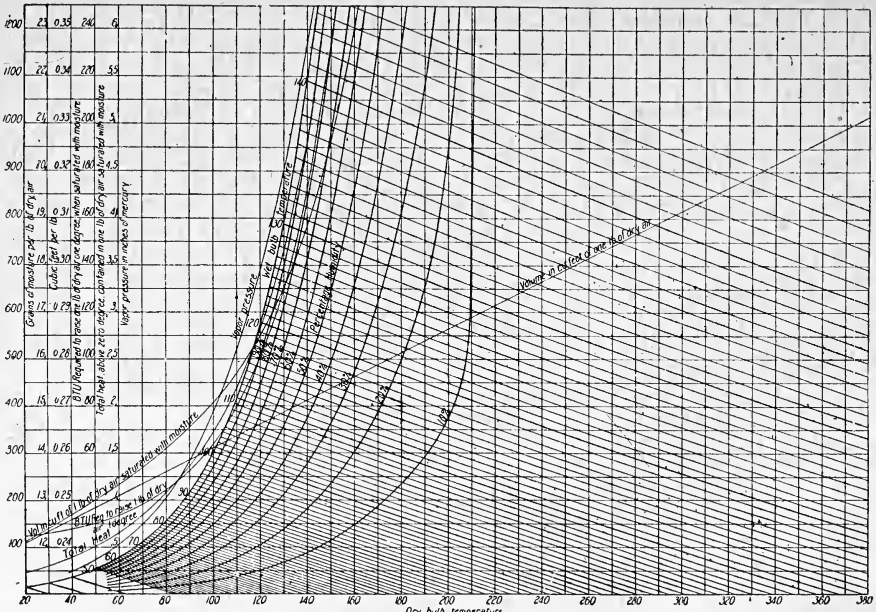


Fig. B. Psychrometric Chart.

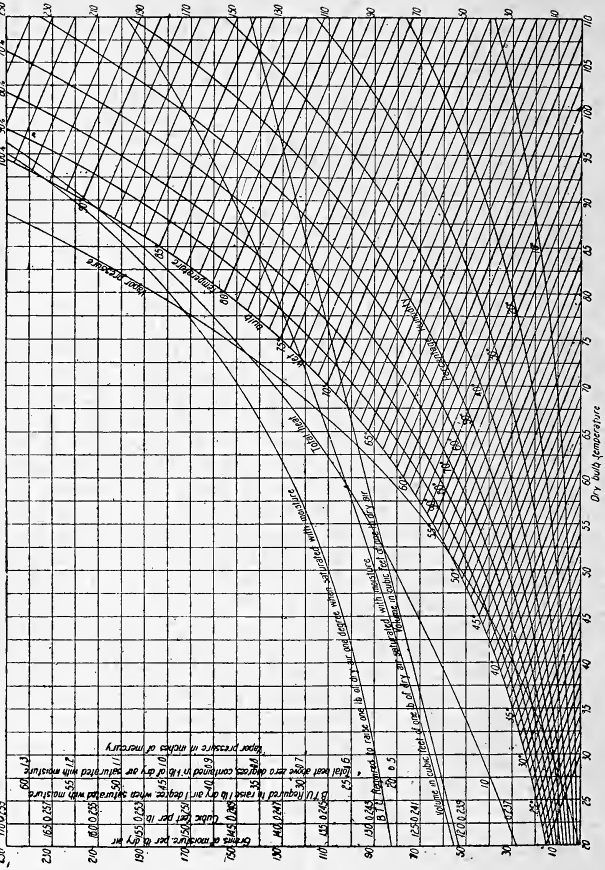


Fig. C. Psychrometric Chart.

TABLE 15.

Fuel Value of American Coals.*

Coal name or locality	Fuel value per pound of coal: B. t. u. calculated	B. t. u. by calorimeter	Theoretical evaporation in lbs. from and at 212 deg. F.
ARKANSAS.			
Spadra Johnson Co.	14,420		14.90
Coal Hill, Johnson Co.		11,812	12.22
Huntington Co.		11,756	12.17
Lignite	9,215		9.54
COLORADO.			
Lignite	13,560		14.04
Lignite, slack	8,500		8.80
ILLINOIS.			
Big Muddy, Jackson Co.		11,781	12.19
Colchester, Slack		9.035	9.35
Giliespie Macoupin Co.		9,739	10.09
Mercer Co.		13,123	13.58
INDIANA.			
Block	14,020		14.50
Cannel	13,097		13.56
IOWA.			
Good Cheer		8,702	9.01
KENTUCKY.			
Caking	14,391		14.89
Cannel	15,198		16.76
Lignite	9,326		9.65
MISSOURI.			
Bevier Mines		9,890	10.24
NEW MEXICO.			
Coal		11,756	12.17
OHIO.			
Briar Hill, Mahoning Co.	13,714		14.20
Hocking Valley	13,414		13.90
PENNSYLVANIA.			
Anthracite	14,199		14.70
Anthracite, pea	12,300		12.73
Pittsburgh (average)		13,104	13.46
Youghiogheney		12,936	13.39
TEXAS.			
Fort Worth		9,450	9.78
Lignite	12,962		13.41
WEST VIRGINIA.			
Pocahontas		14,273	14.71
New River	14,200		14.70

* Sturtevant's "Mechanical Draft."

TABLE 16.

Capacities of Chimneys.*

Inside diameter of lined flue, inches		Maximum sq. ft. of cast iron radiating surface and B. t. u. for a flue of the given diameter and height					
		25 feet high	36 feet high	49 feet high	64 feet high	81 feet high	100 feet high
6	Steam	146	175	204	233	262	291
	Hot water	243	291	340	388	437	485
	B. t. u.	36500	43750	51000	58250	65500	72750
7	Steam	228	273	319	364	410	455
	Hot water	379	455	531	607	683	758
	B. t. u.	57000	68250	79750	91000	102500	113750
8	Steam	327	392	457	523	588	653
	Hot water	544	653	762	871	980	1083
	B. t. u.	81750	98000	114250	130750	147000	163250
9	Steam	445	534	623	712	801	890
	Hot water	742	890	1038	1187	1335	1483
	B. t. u.	111250	133500	155750	178000	200250	222500
10	Steam	582	698	814	930	1047	1163
	Hot water	969	1163	1357	1551	1745	1938
	B. t. u.	145500	174500	203500	232500	261750	290750
12	Steam	909	1090	1272	1454	1636	1817
	Hot water	1514	1817	2120	2423	2726	3028
	B. t. u.	227250	272500	318000	363500	409000	454250
15	Steam	1537	1844	2151	2458	2766	3073
	Hot water	2561	3073	3586	4098	4610	5122
	B. t. u.	384250	461000	537750	614500	691500	768250
18	Steam	2327	2792	3257	3722	4188	4653
	Hot water	3878	4653	5429	6204	6980	7755
	B. t. u.	581750	698000	814250	930500	1047000	1163250

Radiation is calculated at 250 B. t. u. steam, 150 B. t. u. water.

TABLE 17.

Excelsior Double Wall Stack.†

No.	Sizes, Inches			Area Stack, Sq. In.	Collar Diameter, Inches
	Nominal	Inside	Outside		
7	4x11	3x10	3⅝x10⅝	30	8 and 9
8	4x13	3x12	3⅝x12⅝	36	8, 9 and 10
9	4x14	3x13	3⅝x13⅝	39	9 and 10
12	6x13	5x12	5⅝x12⅝	60	9 and 10

* The Model Boiler Manual.

† Excelsior Furnace Co.

TABLE 18.

Equalization of Smoke Flues—Commercial Sizes.*

Inside diameter lined flue	Brick flue not lined well built	Rectangular lined flue outside of tile	Outside iron stack
6	8½x8½		8
7	8½x8½	7x7	9
8	8½x8½	8½x8½	10
9	8½x13	8½x13	11
10	8½x13	8½x13	12
12	13x13	13x13	14
15	13x17	13x18	17
18	17x21½	18x18	20

Round flue tile lining is listed by its inside measurement. Rectangular lining by outside measurement.

TABLE 19.

Dimensions of Registers.

Size of opening, inches	Nominal area of opening, square inches	Effective area of opening, square inches	Tin box size, inches	Extreme dimensions of register face, inches
6 x 10	60	40	6 9/16 x 10 9/16	7 11/16 x 11 11/16
8 x 10	80	53	8⅝ x 10⅝	9¾ x 11¾
8 x 12	96	64	8⅝ x 12⅝	9¾ x 13¾
8 x 15	120	80	8⅝ x 15⅝	9¾ x 16 11/16
9 x 12	108	72	9 11/16 x 12 11/16	10⅞ x 13⅞
9 x 14	126	84	9 11/16 x 14 11/16	10⅞ x 15⅞
10 x 12	120	80	10 11/16 x 12 11/16	11 15/16 x 13 15/16
10 x 14	140	93	10 11/16 x 14 11/16	11 15/16 x 15 15/16
10 x 16	160	107	10 11/16 x 16 11/16	11 15/16 x 17⅞
12 x 15	180	120	12¾ x 15¾	14 1/16 x 17
12 x 19	228	152	12¾ x 19¾	14 1/16 x 21
14 x 22	308	205	14⅞ x 22⅞	16¼ x 24¼
15 x 25	375	250	15⅞ x 25⅞	17¼ x 27¼
16 x 20	320	213	16⅞ x 20⅞	18 5/16 x 22 5/16
16 x 24	384	256	16⅞ x 24⅞	18 5/16 x 26 5/16
20 x 20	400	267	20 15/16 x 20 15/16	22⅜ x 22⅜
20 x 24	480	320	20 15/16 x 24 15/16	22⅜ x 26⅜
20 x 26	520	347	20 15/16 x 26 15/16	22⅜ x 28⅜
21 x 29	609	403	21 15/16 x 29 15/16	23⅜ x 31⅜
27 x 27	729	486	27 15/16 x 27 15/16	29⅜ x 29⅜
27 x 38	1026	684	27 15/16 x 38 15/16	29⅜ x 40⅜
30 x 30	900	600	30 15/16 x 30 15/16	32⅜ x 32⅜

Dimensions of different makes of registers vary slightly. The above are for Tuttle & Bailey manufacture.

* The Model Boiler Manual.

TABLE 20.

Capacities of Warm Air Furnaces of Ordinary Construction in Cubic Feet of Space Heated.*

Divided space			Fire-pot		Undivided space		
+10°	0°	—10°	Diam.	Area	+10°	0°	—10°
12000	10000	8000	18 in.	1.8 sq. ft.	17000	14000	12000
14000	12000	10000	20 "	2.2 "	22000	17000	14000
17000	14000	12000	22 "	2.6 "	26000	22000	17000
22000	18000	14000	24 "	3.1 "	30000	26000	22000
26000	22000	18000	26 "	3.7 "	35000	30000	26000
30000	26000	22000	28 "	4.3 "	40000	35000	30000
35000	30000	26000	30 "	4.9 "	50000	40000	35000

TABLE 21.

Capacities of Hot-Air Pipes and Registers.†

Size of register	Equivalent area in round or leader pipe	Equiv-alent in square or riser pipe	Cubic feet of space on first floor same will heat	Cubic feet on second floor	Cubic feet on third floor
6x8	6 in.	4x8	400	450	500
8x8	7 "	4x10	450	500	560
8x10	8 "	4x10	500	850	880
8x12	8 "	4x11	800	1000	1050
9x12	9 "	4x12	1050	1250	1320
9x14	9 "	4x14	1050	1350	1450
10x12	10 "	4x14	1500	1650	1800
10x14	10 "	6x10	1800	2000	2200
10x16	10 "	6x10	1800	2000	2200
12x14	12 "	6x12	2200	2300	2500
12x15	12 "	6x12	2250	2300	2500
12x17	12 "	6x14	2300	2600	2800
12x19	12 "	6x14	2300	2600	2800
14x18	14 "	6x16	2800	3000	3200
14x20	14 "	6x16	2900	3000	3200
14x22	14 "	8x16	3000	3200	3400
16x20	16 "	8x18	3600	4000	4250
16x24	16 "	8x18	3700	4000	4250
20x24	18 "	10x20	4800	5400	5750
20x26	20 "	10x24	6000	7000	7450

* Federal Furnace League Handbook.
† Kidder's Arch. and B'ld'rs Pocket-Book.

TABLE 22.

Air Heating Capacity of Warm Air Furnaces.*

Fire-pot Diam	Fire-pot Area	Casing Diam.	Total cross sec. area of heat pipes	No. and size of heat pipes that may be supplied
18 in.	1.8 sq. ft.	30″-32″	180 sq. in.	3-9″ or 4-8″
20 "	2.2 "	34″-36″	280 "	2-10″ and 2-9″ or 3-9″ and 2-8″
22 "	2.6 "	36″-40″	360 "	3-10″ and 2-9″ or 4-9″ and 2-8″
24 "	3.1 "	40″-44″	470 "	3-10″, 1-9″ and 2-8″ or 2-10″ and 5-8″
26 "	3.7 "	44″-50″	565 "	5-10″ and 3-9″ or 3-10″, 4-9″ and 2-8″
28 "	4.3 "	48″-56″	650 "	2-12″, 3-10″ and 3-9″ or 5-10″, 3-9″ and 2-8″
30 "	4.9 "	52″-60″	730 "	3-12″, 3-10″ and 3-9″ or 5-10″, 5-9″ and 1-8″

TABLE 23.

Sectional Area (Square Inches) of Vertical Hot Air Flues, Natural Draft, Indirect System.†

Outside temperature 50° F. Flue temperature 90° F.

Sq. ft. cast iron radiation	STEAM First story	STEAM Second story	STEAM Third story	STEAM Fourth story	WATER First story	WATER Second story	WATER Third story	WATER Fourth story
0 to 50	100	75	63	60	75	63	60	60
50 " 75	150	113	94	80	113	94	80	80
75 " 100	200	150	125	100	150	125	100	100
100 " 125	250	188	156	125	188	156	125	125
125 " 150	300	225	188	150	225	188	150	150
150 " 175	350	263	219	175	263	219	175	175
175 " 200	400	300	250	200	300	250	200	200
200 " 225	450	338	281	225	338	281	225	225
225 " 250	500	375	313	250	375	313	250	250
250 " 275	550	413	344	275	413	344	275	275
275 " 300	600	450	375	300	450	375	300	300
300 " 325	650	488	406	325	488	406	325	325
325 " 350	700	525	438	350	525	438	350	350
350 " 375	750	563	469	375	563	469	375	375
375 " 400	800	600	500	400	600	500	400	400
Velocity feet per sec.	2½	4½	5½	6½	1½	2½	4	4
Effective area of register. Factor for	1.00	1.50	1.83	2.17	1.00	1.00	1.33	1.33

* Federal Furnace League Handbook.
† The Model Boiler Manual.

TABLE 24.

Sheet Metal Dimensions and Weights.

Decimal gage	Approximate millimeters	Wt. per sq. ft. in lbs. Iron 480 lbs. per cu. ft.	Wt. per sq. ft. in lbs. Steel 489.6 lbs. per cu. ft.	U. S. Gage numbers
0.002	0.05	0.08	0.082	
0.004	0.10	0.16	0.163	
0.006	0.15	0.24	0.245	38–39
0.008	0.20	0.32	0.326	34–35
0.010	0.25	0.40	0.408	32
0.012	0.30	0.48	0.490	30–31
0.014	0.36	0.56	0.571	29
0.016	0.41	0.64	0.653	27–28
0.018	0.46	0.72	0.734	26–27
0.020	0.51	0.80	0.816	25–26
0.022	0.56	0.88	0.898	25
0.025	0.64	1.00	1.020	24
0.028	0.71	1.12	1.142	23
0.032	0.81	1.28	1.306	21–22
0.036	0.91	1.44	1.469	20–21
0.040	1.02	1.60	1.632	19–20
0.045	1.14	1.80	1.836	18–19
0.050	1.27	2.00	2.040	18
0.055	1.40	2.20	2.244	17
0.060	1.52	2.40	2.448	16–17
0.065	1.65	2.60	2.652	15–16
0.070	1.78	2.80	2.856	15
0.075	1.90	3.00	3.060	14–15
0.080	2.03	3.20	3.264	13–14
0.085	2.16	3.40	3.468	13–14
0.090	2.28	3.60	3.672	13–14
0.095	2.41	3.80	3.876	12–13
0.100	2.54	4.00	4.080	12–13
0.110	2.79	4.40	4.488	12
0.125	3.18	5.00	5.100	11
0.135	3.43	5.40	5.508	10–11
0.150	3.81	6.00	6.120	9–10
0.165	4.19	6.60	6.732	8–9
0.180	4.57	7.20	7.344	7–8
0.200	5.08	8.00	8.160	6–7
0.220	5.59	8.80	8.976	4–5
0.240	6.10	9.60	9.792	3–4
0.250	6.35	10.00	10.200	3

For weights of galvanized iron, multiply weight, black, by:—

No. 28	No. 26	No. 24	No. 22	No. 20	No. 18	No. 16
1.25	1.21	1.16	1.13	1.11	1.08	1.07

TABLE 25.

Weight of Round Galvanized Iron Pipe and Elbows of the Proper Gages for Heating and Ventilating Work.

Gage and weight per sq. ft.	Diam. of pipe	Circumf. of pipe in inches	Area in sq. in.	Weight per running foot	Weight of full elbow
No. 28 0.78	3	9.43	7.1	0.7	0.4
	4	12.57	12.6	1.1	0.9
	5	15.71	19.6	1.2	1.2
	6	18.85	28.3	1.4	1.7
	7	21.99	38.5	1.7	2.3
	8	25.13	50.3	1.9	2.9
No. 26 0.91	9	28.27	63.6	2.4	4.3
	10	31.42	78.5	2.7	5.3
	11	34.56	95.0	2.9	6.4
	12	37.70	113.1	3.2	7.6
	13	40.84	132.7	3.4	8.9
	14	43.98	153.9	3.7	10.4
No. 25 1.03	15	47.12	176.7	4.5	13.5
	16	50.27	201.1	4.7	15.1
	17	53.41	227.0	5.0	17.0
	18	56.55	254.5	5.3	19.1
	19	59.69	283.5	5.6	21.4
	20	62.83	314.2	6.0	23.9
No. 24 1.16	21	65.97	346.4	7.0	29.6
	22	69.12	380.1	7.3	32.3
	23	72.26	415.5	7.7	35.6
	24	75.40	452.4	8.0	38.6
	25	78.54	490.9	8.3	41.7
	26	81.68	530.9	8.7	45.1
No. 22 1.41	27	84.82	572.6	10.9	59.1
	28	87.97	615.7	11.4	64.2
	29	91.11	660.5	11.8	68.6
	30	94.25	706.9	12.2	73.4
	31	97.39	754.8	12.6	78.3
	32	100.53	804.3	13.0	83.4
	33	103.67	855.3	13.5	88.9
	34	106.84	907.9	13.9	94.3
	35	109.96	962.1	14.3	99.9

Gage and weight per sq. ft.	Diam. of pipe	Circumf. of pipe in inches	Area in sq. in.	Weight per running foot	Weight of full elbow
No. 20 1.66	36	113.10	1017.9	17.2	124.4
	37	116.24	1075.2	17.8	131.4
	38	119.38	1134.1	18.2	139.4
	39	122.52	1194.6	18.7	146.0
	40	125.66	1256.6	19.1	152.9
	41	128.81	1320.6	19.6	160.7
	42	131.95	1385.4	20.1	168.6
	43	135.09	1452.2	20.6	176.7
	44	138.23	1520.5	21.0	185.0
	45	141.37	1590.4	21.5	193.4
	46	144.51	1661.9	22.0	202.2
No. 18 2.16	47	147.65	1734.9	29.2	274.3
	48	150.80	1809.6	29.8	286.6
	49	153.94	1885.7	30.4	298.8
	50	157.08	1963.5	31.0	309.9
	51	160.22	2042.8	31.6	322.5
	52	163.36	2123.7	32.2	335.1
	53	166.50	2206.2	33.0	349.7
	54	169.65	2290.2	33.6	463.4
	55	172.79	2375.8	34.4	377.2
	56	175.93	2463.0	34.9	390.7
	57	179.07	2551.8	35.6	405.1
	58	182.21	2642.1	36.1	418.8
	59	185.35	2734.0	36.7	433.1
	60	188.50	2827.4	37.4	448.6
No. 16 2.66	61	191.64	2922.5	46.7	569.7
	62	194.78	3019.1	47.5	589.0
	63	197.92	3117.3	48.3	608.6
	64	201.06	3217.0	49.1	628.5
	66	207.34	3421.2	50.5	666.6
	68	213.63	3631.7	52.1	708.6
	70	219.91	3848.5	53.6	750.4
	72	226.19	4071.5	55.1	793.4

TABLE 26.

Specific Heats, Coefficients of Expansion, Coefficients of Transmission, and Fusing-Points of Solids, Liquids or Gases.*

SUBSTANCE	Specific heats	Coefficient of expansion	Coefficient of transmission	Fusion points, degrees
Antimony	0.0508	.00000602	.00022	815
Copper	0.0951	.00000955	.00404	1949
Gold	0.0324	.00001060	----------	1947
Wrought iron	0.1138	.00000895	.00089	2975
Glass	0.1937	.00000478	.0000008	1832
Cast iron	0.1298	.00000618	.000659	2192
Lead	0.0314	.00001580	.00045	621
Platinum	0.0324	.00000530	----------	3452
Silver	0.0570	.00001060	.00610	1751
Tin	0.0562	.00001500	.00084	446
Steel (soft)	0.1165	.00000600	.00062	2507
Steel (hard)	0.1175	.00000689	.00034	2507
Nickel steel 36%	--------	.00000003	----------	------
Zinc	0.0956	.00001633	.00170	787
Brass	0.0939	.00001043	.00142	1859
Ice	0.5040	.00000375	.000024	32
Sulphur	0.2026	.00006413	----------	------
Charcoal	0.2410	.00007860	.000002	------
Aluminum	0.1970	.00002313	.00203	1213
Phosphorus	0.1887	.00012530	----------	------
Water	1.0000	.00008806	.000008	------
Mercury	0.0333	.00003333	.00011	------
Alcohol (absolute)	0.7000	.00015151	.000002	------

	Constant pressure	Constant volume	Coefficient of cubical expansion at 1 atmos.		
Air	0.23751	0.16847	.003671	.0000015	------
Oxygen	0.21751	0.15507	.003674	.0000012	------
Hydrogen	3.40900	2.41226	.003669	.0000012	------
Nitrogen	0.24380	0.17273	.003668	.0000012	------
Superheated steam	0.4805	0.346	.003726	--------	------
Carbonic acid	0.2170	0.1535	--------	.00000122	------

* Kent and Suplee.

TABLE 27.

Pressure in Ounces, per Square Inch, Corresponding to Various Heads of Water, in Inches.*

Head in inches	Decimal parts of an inch									
	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0	------	.06	.12	.17	.23	.29	.35	.40	.46	.52
1	.58	.63	.69	.75	.81	.87	.93	.98	1.04	1.09
2	1.16	1.21	1.27	1.33	1.39	1.44	1.50	1.56	1.62	1.67
3	1.73	1.79	1.85	1.91	1.96	2.02	2.08	2.14	2.19	2.25
4	2.31	2.37	2.42	2.48	2.54	2.60	2.66	2.72	2.77	2.83
5	2.89	2.94	3.00	3.06	3.12	3.18	3.24	3.29	3.35	3.41
6	3.47	3.52	3.58	3.64	3.70	3.75	3.81	3.87	3.92	3.98
7	4.04	4.10	4.16	4.22	4.28	4.33	4.39	4.45	4.50	4.56
8	4.62	4.67	4.73	4.79	4.85	4.91	4.97	5.03	5.08	5.14
9	5.20	5.26	5.31	5.37	5.42	5.48	5.54	5.60	5.66	5.72

TABLE 28.

Height of Water Column, in Inches, Corresponding to Pressures, in Ounces, per Square Inch.*

Pressure in ounces per square inch	Decimal parts of an ounce									
	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0	------	.17	.35	.52	.69	.87	1.04	1.21	1.38	1.56
1	1.73	1.90	2.08	2.25	2.42	2.60	2.77	2.94	3.11	3.29
2	3.46	3.63	3.81	3.98	4.15	4.33	4.50	4.67	4.84	5.01
3	5.19	5.36	5.54	5.71	5.88	6.06	6.23	6.40	6.57	6.75
4	6.92	7.09	7.27	7.44	7.61	7.79	7.96	8.13	8.30	8.48
5	8.65	8.82	9.00	9.17	9.34	9.52	9.69	9.86	10.03	10.21
6	10.38	10.55	10.73	10.90	11.07	11.26	11.43	11.60	11.77	11.95
7	12.11	12.28	12.46	12.63	12.80	12.97	13.15	13.32	13.49	13.67
8	13.84	14.01	14.19	14.36	14.53	14.71	14.88	15.05	15.22	15.40
9	15.57	15.74	15.92	16.09	16.26	16.45	16.62	16.76	16.96	17.14

* Suplee's M. E. Reference Book.

TABLE 29. **Wrought Iron and Steel, Steam, Gas and Water Pipe.**

Diameter: Nominal internal inches	Diameter: Actual external inches	Diameter: Approx. internal inches	Nominal Thickness—inches	Circumference: External inches	Circumference: Internal inches	Transverse Areas: External sq. inches	Transverse Areas: Internal sq. inches	Transverse Areas: Metal sq. inches	Length of pipe per sq. ft. of External surface, ft.	Length of pipe per sq. ft. of Internal surface, ft.	Feet of pipe containing one cubic foot	Normal weight per foot	No. of threads	Weight of water per lineal foot
⅛	.405	.270	.068	1.272	.845	.129	.0568	.0720	9.440	14.150	2513.0	.241	27	.024
¼	.540	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.490	1383.3	.420	18	.044
⅜	.675	.494	.091	2.121	1.549	.358	.1909	.1669	5.657	7.760	751.2	.559	18	.082
½	.840	.623	.109	2.629	1.954	.554	.3039	.2503	4.547	6.150	472.4	.837	14	.132
¾	1.050	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.0	1.115	14	.230
1	1.315	1.048	.134	4.131	3.289	1.358	.8609	.4972	2.904	3.645	166.9	1.668	11½	.373
1¼	1.660	1.380	.140	5.215	4.335	2.164	1.496	.6685	2.301	2.768	96.25	2.244	11½	.648
1½	1.900	1.611	.145	5.969	5.058	2.835	2.038	.7995	2.010	2.371	70.66	2.678	11½	.883
2	2.375	2.067	.154	7.461	6.494	4.430	3.356	1.074	1.608	1.848	42.91	3.609	11½	1.454
2½	2.875	2.468	.204	9.032	7.750	6.492	4.780	1.712	1.328	1.547	30.10	5.739	8	2.072
3	3.500	3.067	.217	10.996	9.632	9.621	7.388	2.238	1.091	1.245	19.50	7.536	8	3.202
3½	4.000	3.548	.226	12.566	11.146	12.566	9.887	2.680	.955	1.077	14.57	9.001	8	4.285
4	4.500	4.026	.237	14.137	12.648	15.904	12.730	3.175	.849	.949	11.31	10.665	8	5.517
4½	5.000	4.508	.246	15.708	14.162	19.635	15.961	3.675	.764	.848	9.02	12.340	8	6.908
5	5.563	5.045	.259	17.477	15.849	24.306	19.985	4.321	.687	.757	7.20	14.502	8	8.668
6	6.625	6.065	.280	20.813	19.054	34.472	28.886	5.586	.577	.630	4.98	18.762	8	12.521
7	7.625	7.023	.301	23.955	22.063	45.664	38.743	6.921	.501	.544	3.72	23.271	8	16.790
8	8.625	7.982	.322	27.096	25.073	58.426	50.021	8.405	.443	.478	2.88	28.177	8	21.688
9	9.625	8.937	.344	30.238	28.076	72.760	62.722	10.040	.397	.427	2.29	33.701	8	27.580
10	10.750	10.019	.366	33.772	31.472	90.763	78.882	11.940	.355	.381	1.82	40.065	8	34.171
11	11.750	11.000	.375	36.914	34.558	108.434	95.034	13.401	.325	.348	1.51	45.950	8	41.189
12	12.750	12.000	.375	40.055	37.700	127.677	113.098	14.590	.299	.319	1.27	48.985	8	49.017
13	14.000	13.250	.375	43.980	41.620	153.940	137.880	16.060	.270	.290	1.04	54.000	8	59.762
14	15.000	14.250	.375	47.120	44.760	176.710	159.480	17.230	.250	.270	.90	58.000	8	69.125
15	16.000	15.400	.280	50.260	48.480	201.060	187.040	14.020	.240	.250	.77	62.000	8	81.070
16	17.000	16.400	.300	53.410	51.520	226.980	211.240	15.740	.230	.230	.68	49.188	8	91.559
17	18.000	17.300	.340	56.550	54.410	254.470	235.610	18.860	.210	.220	.61	58.920	8	102.122

TABLE 30.

Expansion of Wrought-Iron Pipe on the Application of Heat.*

Temp. air when pipe is fitted	Increase in length in inches per 100 feet when heated to							
Deg. F.	160	180	200	212	220	228	240	274
0	1.28	1.44	1.60	1.70	1.76	1.82	1.92	2.19
32	1.02	1.18	1.34	1.44	1.50	1.57	1.66	1.94
50	.88	1.04	1.20	1.30	1.36	1.42	1.52	1.79
70	.72	.88	1.04	1.14	1.20	1.26	1.36	1.63

TABLE 31.

Tapping List of Direct Radiators.†

STEAM.

ONE-PIPE WORK		TWO-PIPE WORK	
Radiator area square feet	Tapping diameter—inches	Radiator area square feet	Tapping diameter—inches
0 — 24	1	0 — 48	1 x ¾
24 — 60	1¼	48 — 96	1¼x1
60 — 100	1½	96 and above	1½x1¼
100 and above	2		

WATER.

Tapped for supply and return.

Radiator area square feet	Tapping diameter inches
0 — 40	1
40 — 72	1¼
72 and above	1½

* Holland Heating Manual.

† American Radiator Co.

TABLE 32.

Pipe Equalization. (See also Table 21)

This table shows the relation of the combined area of small round warm air ducts or pipes to the area of one large main duct.

The bold figures at the top of the column represent the diameters of the small pipes or ducts; those in the left-hand vertical columns are the diameters of the main pipes. The small figures show the number of small pipes that each main duct will supply.

Example.—To supply sixteen 10-inch pipes: Refer to column having 10 at top; follow down to small figure 16, thence left on the horizontal line of the bold-face figure in the outside column, and we find that one 30-inch main will supply air for the sixteen 10 - inch pipes.

	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**	**29**	**30**
1																														
2	5.7																													
3	16	2.7																												
4	32	5.7	2.3																											
5	56	9.7	3.6	1.8																										
6	88	16	5.7	2.8	1.6																									
7	129	23	8.3	4.1	2.3	1.5																								
8	180	32	12	5.7	3.2	2.1	1.4																							
9	244	42	16	7.6	4.3	2.8	1.9	1.3																						
10	317	56	20	9.9	5.7	3.6	2.4	1.7	1.3																					
11	402	71	26	12	7.0	4.5	3.1	2.2	1.7	1.3																				
12	501	88	32	16	9.0	5.7	3.8	2.8	2.0	1.6	1.2																			
13	613	107	39	19	11	6.9	4.7	3.4	2.5	1.9	1.5	1.2																		
14	737	129	47	23	13	8.3	5.7	4.1	3.0	2.3	1.8	1.5	1.2																	
15	876	152	56	27	16	9.9	6.7	4.8	3.6	2.8	2.2	1.8	1.4	1.2																
16	1026	180	65	32	18	11	7.9	5.7	4.2	3.2	2.6	2.1	1.7	1.4	1.2															
17	1197	208	76	37	21	13	9.2	6.6	4.9	3.8	2.9	2.4	2.0	1.6	1.4	1.2														
18	1375	239	88	43	24	16	10	7.7	5.7	4.3	3.4	2.8	2.3	1.9	1.6	1.3	1.2													
19	1580	275	100	49	28	18	12	8.8	6.5	5.0	3.9	3.2	2.6	2.2	1.8	1.5	1.3	1.2												
20	1775	312	114	56	32	20	14	9.9	7.4	5.7	4.5	3.6	2.9	2.5	2.1	1.7	1.5	1.3	1.1											
21	1985	345	130	61	35	22	15	10	8.4	6.3	5.2	4.1	3.2	2.6	2.3	2.0	1.6	1.4	1.2	1.1										
22	2250	398	145	71	41	26	18	13	9.3	7.2	5.7	4.5	3.7	3.1	2.6	2.2	1.9	1.7	1.4	1.3	1.1									
23	2525	460	160	77	47	29	20	14	10	7.8	6.2	5.0	4.1	3.4	2.8	2.4	2.1	1.8	1.6	1.4	1.2	1.1								
24	2800	493	180	88	50	32	22	16	12	8.9	7.6	5.7	4.6	3.8	3.2	2.9	2.4	2.1	1.8	1.6	1.3	1.2	1.1							
25	3060	543	202	97	55	35	24	17	13	10	8.0	6.4	5.2	4.2	3.5	3.1	2.6	2.2	2.0	1.7	1.5	1.3	1.2	1.1						
26	3425	590	219	108	62	39	27	19	14	11	8.6	6.9	5.7	4.7	4.0	3.4	2.9	2.5	2.2	1.9	1.7	1.5	1.3	1.2	1.1					
27	3738	677	243	121	68	43	29	21	15	12	9.6	7.5	6.1	5.1	4.3	3.6	3.1	2.7	2.4	2.1	1.9	1.6	1.5	1.3	1.2	1.1				
28	4100	725	265	129	74	48	32	23	17	13	10	8.3	6.8	5.7	4.8	4.1	3.5	3.0	2.6	2.3	2.1	1.8	1.6	1.5	1.3	1.2	1.1			
29	4440	800	289	141	79	52	35	25	19	14	11	9.1	1.5	6.2	5.2	4.4	3.8	3.3	2.9	2.5	2.2	2.0	1.8	1.6	1.4	1.3	1.2	1.1		
30	4898	864	315	151	88	56	38	28	20	16	12	9.9	8.0	6.7	5.7	4.7	4.1	3.6	3.0	2.6	2.4	2.2	1.9	1.7	1.5	1.4	1.3	1.2	1.1	
31	5312	920	344	168	96	59	40	30	22	17	13	10	8.9	7.4	6.1	5.1	4.5	3.9	3.4	3.0	2.7	2.4	2.1	1.9	1.7	1.5	1.4	1.3	1.2	1.1
32	5631	1070	374	184	103	63	42	32	23	18	14	11	9.3	8.0	6.5	5.7	4.8	4.2	3.7	3.2	2.8	2.5	2.3	2.0	1.8	1.6	1.5	1.4	1.3	1.1
33	6154	1140	401	196	109	70	45	35	25	19	15	12	10	8.5	7.1	6.0	5.2	4.4	3.9	3.5	3.0	2.7	2.4	2.2	2.0	1.8	1.6	1.5	1.3	1.2
34	6675	1208	433	212	119	76	51	37	27	21	16	13	11	9.1	7.6	6.4	5.5	4.8	4.2	3.7	3.3	2.9	2.6	2.4	2.1	1.9	1.7	1.6	1.5	1.3
35	7075	1280	470	229	127	82	56	40	30	23	18	14	12	10	8.4	7.1	6.1	5.3	4.6	4.0	3.5	3.1	2.8	2.5	2.3	2.1	1.9	1.7	1.5	1.4
36	7735	1355	497	242	138	88	60	43	32	25	19	16	13	11	8.9	7.6	6.5	5.7	5.0	4.3	3.7	3.4	3.0	2.7	2.4	2.2	2.0	1.9	1.7	1.6
37	8265	1435	537	260	146	94	64	45	33	26	20	16	13	11	9.3	7.9	6.9	6.0	5.3	4.6	4.0	3.6	3.2	2.9	2.6	2.3	2.1	1.9	1.8	1.6
38	8715	1625	575	279	157	100	68	49	37	29	22	18	14	12	10	8.6	7.4	6.3	5.5	4.8	4.3	3.8	3.4	3.1	2.8	2.5	2.3	2.1	1.9	1.8

TABLE 33.

Sizes of Hot-Water Mains.

Open Tank System.

Assumed Length 100 feet.†

Pipe diam. inches	Capacity, square feet of direct radiation		
	Two-pipe* up feed	One-pipe up feed	Attic main
1¼	100	50	150
1½	150	75	225
2	275	125	375
2½	375	225	540
3	600	400	900
3½	800	500	1300
4	1100	700	1800
5	1900	1200	3200
6	3000	2000	5000
7	4500	3000	7200
8	6000	4000	10000

† For mains over 100′ reduce capacity in the ratio of $\sqrt{\frac{100}{\text{length}}}$

*Mains for *indirect radiation* should have a rated capacity approximating 66 per cent. of the values in this column.

TABLE 34.

Sizes of Hot-Water Branches and Risers.

Open Tank System.

Pipe diam. inches	Up Feed				Down feed from attic not exceeding four floors
	First Floor	Second Floor	Third Floor	Fourth Floor	
1	50	65	75	85	75
1¼	90	110	130	145	125
1½	125	160	190	215	200
2	225	300	350	375	350
2½	325	425	510	580	600
3	500	600	700	800	900

Take first floor supply branches from top of main. Risers above first floor at 45°.

TABLE 35.

Hot-Water Radiator Tappings.

Open Tank System.

Size of Radiator	Supply and Return
Up to 40 sq. ft.	1 x 1
40 to 72 sq. ft.	1¼ x 1¼
Above 72 sq. ft.	1½ x 1½

TABLE 36.

Honeywell System. Pipe Sizes.

The *area of the main* must equal or exceed slightly the combined area of the valves it is to supply.

Riser Sizes and Square Feet of Radiation.			
Pipe size, inches	First Floor	Second Floor	Third Floor
½	Up to 30	Up to 40	Up to 50
¾	30 to 60	40 to 100	50 to 125
1	60 to 100	Over 100	Over 125

The valve on the radiator at the end of the main should generally be made one size larger than the list.

TABLE 37.

Gravity Hot-Water Heating. Approximate Capacities of Mains and Risers for Range from 180 to 150 Deg. Fahr.‡

Capacities (including losses in transit) are in 1000 B. t. u. per hour and allow for average resistance of boilers, radiators and piping. For sq. ft. of radiating surface supplied (160 B. t. u. per sq. ft.), multiply the tabular figures by 6.25.

MAINS.

Length,* feet	Height,* feet	Diameter of main, in.											
		1¼	1½	2	2½	3	3½	4	4½	5	6	7	8
		Capacity in 1000 B. t. u.											
100	7	15	22	40	60	98	133	188	240	315	480	675	900
200	8	12.5	18	32	50	82	114	157	206	270	415	590	800
300	9	11	16	29	45	75	106	144	190	250	385	550	740
400	10	10	15	27	42	70	100	135	180	238	367	520	700

RISERS.

Height,† feet	Diameter of riser, in.						Height,† feet	Diameter of riser, in.					
	¾	1	1¼	1½	2	2½		½	¾	1	1¼	1½	2
	Capacity in 1000 B. t. u.							Capacity in 1000 B. t. u.					
10	4.0	7.5	15.0	22.0	42	67	30	3.4	7.1	13.1	26	38	74
15	5.0	9.2	18.7	27.4	52	82	40	4.0	8.2	15.2	31	45	86
20	5.8	10.6	21.7	31.8	60	95	50	4.5	9.2	17.0	35	51	97
25	6.4	11.8	24.2	35.5	67	106	60	4.9	10.1	18.5	38	56	107

* The length and mean height above boiler are those of the circuit for the most distant radiator in lowest location.

† The mean height above boiler is that of the circuit in question. This table is for a circuit 200 ft. long. For other lengths allow about in proportion as given above for mains.

‡ Marks—M. E. Handbook.

TABLE 38.

Capacities of Steam and Return Pipes.*

Unwins formula. Friction, 1 oz. per 100 ft. This will probably be doubled by friction of moisture, condensation, fittings, etc. Friction of gravity returns, 2 oz. per 100 ft. including fittings. Condensation .3 lbs. per sq. ft. per hr. as shown by report of N. D. H. A. Research Committee, May, 1920.

	½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	
SUPPLY PIPING																
Steam Main and Down Feed Riser, all systems; Up Feed Riser and Rad. Conn., two pipe		20	40	75	150	300	500	900	1500	2000	2800	3600	6000	9000	13000	9—18000
Up Feed Riser, one pipe			40	75	125	280	460	670	900	1200	1500	1860	2700			10—23000
Radiator Conn., one pipe			24	60	100	200										12—37000
RETURN PIPING																14—55000
Wet Ret. and Wet Drips, all systems		800	1600	3000	6000	12000	20000	36000	60000	80000						16—78000
Mechanical Vacuum System																
" " Main Return		600	1200	2400	4800	9000	15000	23000	37000	55000	78000					
" " Return Riser		400	600	1200	2400	4800										
" " Ret. Rad. Conn.	100	200	400	600												
" " Drip					¾	¾	¾	1	1	1	1¼	1¼	1¼			
Gravity Air—Return System																
" " Dry Ret. Main			500	1000	2000	4000	7000	12000	20000	27000	37000	48000	80000			
" " Return Riser		200	400	800	1500	3000	5000									
" " Ret. Rad. Conn.	50	100	200	400												
" " Drip				¾	¾	¾	1	1	1	1¼	1¼	1¼	1¼			
Two-Pipe Gravity																
" " Dry Ret. Main			75	150	300	500	1500	2800	6000	13000	18000	23000	37000	55000	78000	
" " Riser to Wet Ret.		150	300	500	900	2000	3600	6000								
" " Riser to Dry Ret.		40	75	150	300	500	1500	2800	6000							
" " Return Rad. Conn.		40	75	150	300											
" " Drip to Wet Ret.			¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½			
" " Drip to Dry Ret.			¾	¾	1	1	1¼	1¼	1½	1½	2	2	2½			
One-Pipe Gravity																
" " Dry Drip Main			75	150	300	500	1500	2800	6000	13000	18000	23000	37000	55000	78000	
" " Wet Drip Main		800	1600	3000	6000	12000	20000	36000	60000	80000						
" " Drip to Wet Ret.			¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½			
" " Drip to Dry Ret.			¾	¾	1	1	1¼	1¼	1½	1½	2	2	2½			

*J. A. Donnelly.

TABLE 39. **Sizes of Steam Mains.***

Pounds of Steam Delivered per minute. See Equation 129.

Diam. in inches	LENGTH IN FEET															
	50	100	175	250	325	400	475	550	700	850	1000	1150	1300	1450	1600	1750
3	65.17	46.0	35.0	29.2	25.6	23.0	21.2	19.6	17.4	15.8	14.5	13.6	12.7	12.10	11.5	11.0
3½	98.3	69.5	52.7	44.6	38.6	34.7	31.9	29.6	26.3	23.8	22.0	20.5	19.2	18.2	17.4	16.6
4	138.1	97.6	73.8	61.8	54.2	48.8	44.8	41.5	36.7	33.5	30.8	28.7	27.1	25.6	24.4	23.3
4½	187.9	132.9	102.0	84.1	73.8	66.45	61.2	56.6	50.1	45.6	42.0	39.2	36.75	34.75	33.2	31.8
5	255.6	180.7	136.2	114.3	102.2	90.3	83.3	76.9	68.2	62.1	57.1	53.2	50.0	47.2	45.2	43.2
6	419.4	296.5	224.1	187.4	164.2	148.2	136.1	125.8	111.8	101.8	93.7	87.2	82.2	77.6	74.1	71.0
7	618	437	330	276	242	218	200	186	165	150	138	129	121	115	109	104
8	890	624	472	394	346	312	286	266	236	214	197	184	173	164	156	149
9	1206	853	644	539	473	426	391	363	322	292	269	251	236	224	213	204
10	1592	1126	851	712	624	563	516	480	425	386	356	332	312	295	281	269
11	2046	1447	1093	915	803	723	664	617	546	496	457	426	401	380	361	346
12	2575	1887	1428	1192	1044	943	852	803	714	648	598	557	523	494	472	451
13	3165	2238	1692	1415	1242	1119	1027	954	846	767	707	660	620	588	559	535

Diam. in inches	LENGTH IN FEET															
	100	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000	2200
14	2714	1920	1567	1357	1213	1108	1025	959	904	858	783	725	678	639	606	578
15	3250	2300	1873	1625	1453	1327	1228	1149	1083	1028	938	868	812	766	726	693
16	4000	2830	2315	2000	1785	1638	1513	1413	1333	1268	1153	1072	1000	945	895	854
17	4500	3210	2635	2250	2021	1841	1702	1591	1500	1424	1302	1200	1124	1061	1006	962
18	5211	3685	3008	2605	2330	2128	1970	1843	1737	1648	1504	1393	1302	1228	1165	1111
19	5992	4237	3459	2996	2680	2447	2265	2119	1997	1895	1730	1602	1498	1412	1340	1278
20	6839	4835	3948	3419	3059	2793	2585	2418	2279	2163	1974	1828	1709	1612	1529	1458
22	8743	6183	5048	4371	3910	3570	3278	3093	2914	2765	2524	2337	2185	2061	1955	1864
24	11308	7990	6535	5650	5065	4522	4435	3995	3769	3580	3271	3023	2820	2665	2535	2415

* Trans. A. S. M. E. Vol. XX, page 358.

TABLE 40.

Sizes for Steam Supply and Return Lines.†

Pipe Sizes	½	¾	1	1¼	1½	2	2½	3	3½
Supply mains, all systems; downfeed risers, all systems	----	----	50	100	175	350	600	1000	1500
Upfeed risers, one-pipe system	----	----	----	50	100	200	300	500	700
Dry return lines, two-pipe and vapor systems	----	50	150	300	900	2000	3800	6000	10000
Wet return lines	----	----	2000	3800	6000	13000	23000	37000	55000
Vacuum return lines	100	400	800	1500	3000	6000	10000	18000	30000

Pipe Sizes	4	5	6	8	10	12	14	16
Supply mains, all systems; downfeed risers, all systems	2000	3800	6000	13000	23000	35000	55000	78000
Upfeed risers,* one-pipe system	800	1300	1800	3000	-----	-----	-----	-----
Dry return lines, two-pipe and vapor systems	13000	23000	37000	78000	-----	-----	-----	-----
Wet return lines	78000	-----	-----	-----	-----	-----	-----	-----
Vacuum return lines	40000	65000	-----	-----	-----	-----	-----	-----

* Which carry condensation from radiators.

TABLE 41.

Sizes of Radiator Connections.†

One-pipe radiators			Two-pipe radiators		
Size of Radiator, square feet	Radiator connection	Horizontal branch	Size of Radiator, square feet	Size of supply connection	Size of return connection
20	1	1	48	1	¾
24	1	1¼	96	1¼	1
40	1¼	1¼	over 96	1½	1¼
60	1¼	1½			
80	1½	1½			
100	1½	2			
200	2	2			

† Allen and Walker.

TABLE 42.

Loss of Head by Friction of Pipes.*

Loss of head by friction in each 100 feet in length of different diameters of pipe when discharging the following quantities of water per minute.

VELOCITY IN FEET PER SECOND	INSIDE DIAMETER OF PIPE IN INCHES															
	1		2		3		4		5		6		7		8	
	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute	Loss of head in feet	Cubic feet per minute
2.0	2.37	.65	1.185	2.62	.791	5.89	.593	10.4	.474	16.3	.395	23.5	.338	32.0	.296	41.9
3.0	4.89	.99	2.44	3.92	1.62	8.83	1.22	15.7	9.78	24.5	.815	35.3	.698	48.1	.611	62.8
4.0	8.20	1.32	4.10	5.23	2.73	11.80	2.05	20.9	1.64	32.7	1.37	47.1	1.175	64.1	1.027	83.7
5.0	12.33	1.65	6.17	6.54	4.11	14.70	3.08	26.2	2.46	40.9	2.05	58.9	1.76	80.2	1.54	105.0
6.0	17.23	1.98	8.61	7.85	5.74	17.70	4.31	31.4	3.45	49.1	2.87	70.7	2.46	96.2	2.15	125.
7.0	22.89	2.31	11.45	9.16	7.62	20.6	5.72	36.6	4.57	57.2	3.81	82.4	3.26	112.0	2.85	146.
	9		10		11		12		14		16		18		20	
2.0	.264	53.	.237	65.4	.216	79.2	.198	94.2	.169	128	.147	167	.132	212	.119	262
3.0	.544	79.5	.488	98.2	.444	119	.407	141.	.349	192	.306	251	.271	318	.245	393
4.0	.913	106.	.822	131	.747	158	.685	188.	.587	256	.513	335	.456	424	.410	523
5.0	1.37	132.	1.23	163	1.122	198	1.028	235.	.881	321	.770	419	.685	530	.617	654
6.0	1.92	159.	1.71	196	1.56	237	1.43	283.	1.229	385	1.076	502	.957	636	.861	785
7.0	2.52	185.	2.28	229	2.07	277	1.91	330.	1.63	449	1.43	586	1.27	742	1.143	916

* Dayton Hydraulic Co. Catalog.

TABLE 43.

Comparative Sizes of Steam Mains and Returns for Gravity and Mechanical Vacuum Systems.

Size of supply pipe	Size of return		Size of supply pipe	Size of return	
	Gravity	Vacuum		Gravity	Vacuum
¾	¾	½	4	2½	1½
1	¾	½	4½	2½	1½
1¼	1	½	5	3	2
1½	1¼	¾	6	3½	2½
2	1½	¾	8	4½	3½
2½	2	1	10	6	4
3	2	1¼	12	6	4½
3½	2½	1¼	14	7	5

Note.—For short runs of piping where the friction is not a serious matter the above table will work out satisfactorily. These sizes are only approximate and should be used with caution.

TABLE 44.

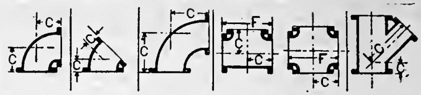
Expansion Tanks—Dimensions and Capacities.*

Size in inches	Capacity gallons	Sq. ft. of radiation
9x20	5½	150
10x20	8	250
12x20	10	350
12x24	12	450
12x30	15	550
12x36	18	650
14x30	20	700
14x36	24	850
16x30	26	900
16x36	32	1250
16x48	42	1750
18x60	66	2750
20x60	82	4500
22x60	100	6000
24x60	122	7500

* The Model Boiler Manual.

TABLE 45.

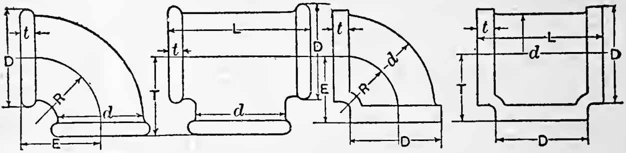
Sizes of Flanged Fittings.



Pipe size in inches	All fittings and flanges: Diameter of flange	Thickness of flange	No. of bolts	Diameter of bolt circle	Size of bolts	90° elbow: Center to face "C"	45° elbow: Center to face "C"	Long turn elbow: Center to face "C"	Tee: Center to face "C"	Cross: Face to face "F"	Lateral: Center to face "C"	Lateral: Center to face short end "C"
4	9	15/16	8	7½	¾	6½	4	10	6½	13	12	3
6	11	1	8	9½	¾	8	5	13	8	16	14½	3½
8	13½	1⅛	8	11¾	¾	9	6	16	9	18	17½	4½
10	16	1 3/16	12	14¼	⅞	11	7	20	11	22	20½	5
12	19	1¼	12	17	⅞	12	7½	22	12	24	24½	5½
14	21	1⅜	14	18¾	1	14	7½	24	14	28	27	6
16	23½	1 7/16	16	21¼	1	15	8	28	15	30	30	6½
20	27½	1 11/16	20	25	1⅛	18	9½	32	18	36	35	8
24	32	1⅞	20	29½	1⅛	22	11	36	22	44	40½	9

TABLE 46.

Dimensions of Ells and Tees for Wrought Iron Pipe.



Size	E	R	D	d	t	L	T
⅛	⅝	9/16	13/16	9/16	3/16	1-¼	⅝
¼	¾	⅝	1-	¾	3/16	1-½	¾
⅜	⅞	¾	1-⅛	⅞	¼	1-¾	⅞
½	1-⅛	⅞	1-¼	1-1/16	¼	2-¼	1-⅛
¾	1-⅜	1-1/16	1-9/16	1-5/16	5/16	2-¾	1-⅜
1-	1-9/16	1-¼	1-⅞	1-⅝	5/16	3-⅛	1-9/16
1-¼	1-⅞	1-½	2-¼	2-	5/16	3-¾	1-⅞
1-½	2-	1-⅝	2-½	2-¼	⅜	4-	2-
2-	2-⅜	2-⅛	3-⅜	2-⅞	¾	4-¾	2-⅜
2-½	2-⅞	2-½	4-	3-½	¾	5-¾	2-⅞
3-	3-⅜	2-¾	4-⅝	4-	⅞	6-¾	3-⅜
3-½	3-⅝	3-⅛	5-¼	4-⅝	⅞	7-¼	3-⅝
4-	4-	3-⅜	5-⅞	5-¼	1-	8-	4-
4-½	4-⅜	4-	6-⅛	6-	1-	8-¾	4-⅜
5-	4-⅝	4-⅛	6-½	6-⅞	1-⅛	9-½	4-¾
6-	5-¼	4-⅞	8-½	7-⅞	1-⅛	11-	5-½

TABLE 47.

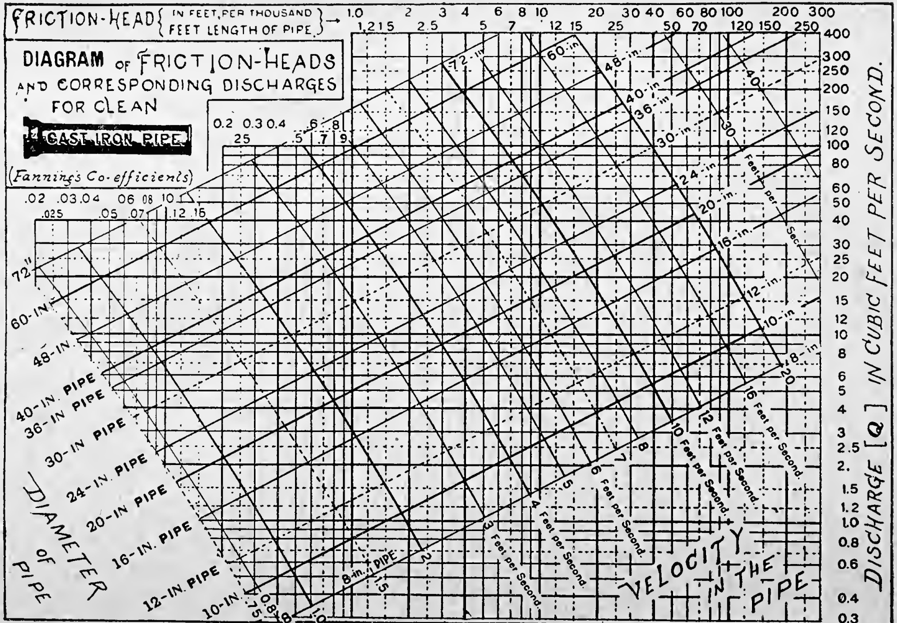
Loss of Pressure in Pipes 100 Feet Long in Ounces per Square Inch when Delivering Air at the Velocities Given.

Velocity in ft. per min.	Diameter of pipe in inches										
	1	2	3	4	6	8	10	12	14	16	18
300	0.100	0.050	0.033	0.025	0.017	0.012	0.010	0.008	0.007	0.006	0.006
400	0.178	0.088	0.059	0.044	0.030	0.022	0.018	0.015	0.013	0.011	0.010
600	0.400	0.200	0.133	0.100	0.067	0.050	0.040	0.033	0.029	0.025	0.022
800	0.711	0.356	0.237	0.178	0.119	0.089	0.071	0.059	0.051	0.044	0.040
1000	1.111	0.556	0.370	0.278	0.185	0.139	0.111	0.092	0.079	0.069	0.062
1200	1.600	0.800	0.533	0.400	0.267	0.200	0.160	0.133	0.114	0.100	0.089
1500	2.500	1.250	0.833	0.625	0.417	0.312	0.250	0.208	0.179	0.156	0.139
1800	3.600	1.800	1.200	0.900	0.600	0.450	0.360	0.300	0.257	0.225	0.200
2400	6.400	3.200	2.133	1.600	1.067	0.800	0.640	0.533	0.457	0.400	0.356
	20	24	28	32	36	40	44	48	52	56	60
300	0.005	0.004	0.004	0.003	0.003	0.002	0.002	0.002	0.002	0.002	0.002
400	0.009	0.007	0.006	0.006	0.005	0.004	0.004	0.004	0.003	0.003	0.003
600	0.020	0.017	0.014	0.012	0.011	0.010	0.009	0.008	0.008	0.007	0.007
800	0.036	0.029	0.025	0.022	0.020	0.018	0.016	0.015	0.014	0.013	0.012
1000	0.056	0.046	0.040	0.035	0.031	0.028	0.025	0.023	0.021	0.020	0.019
1200	0.080	0.067	0.057	0.050	0.044	0.040	0.036	0.033	0.031	0.029	0.027
1500	0.125	0.104	0.089	0.078	0.069	0.062	0.057	0.052	0.048	0.045	0.042
1800	0.180	0.167	0.129	0.112	0.100	0.090	0.082	0.075	0.069	0.064	0.060
2400	0.320	0.313	0.239	0.200	0.178	0.160	0.145	0.133	0.123	0.119	0.107

Diagrams for Pipe Sizes and Friction Heads.

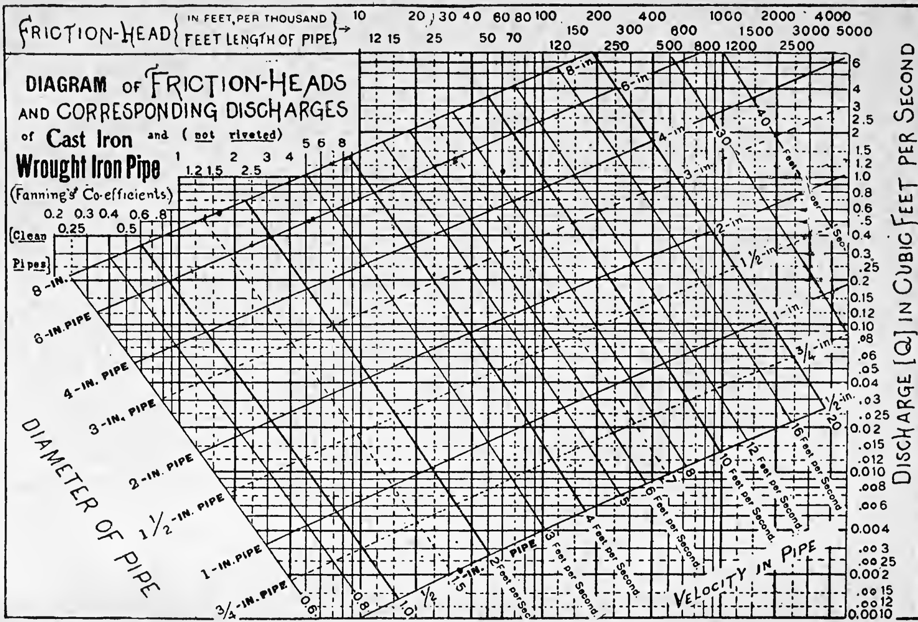
To illustrate the use of the two following diagrams, apply to the pipe line, *B*, *C*, Art. 147. First, let l = 1500 feet, d = 8 inches and v = 5 feet per second. Trace along the velocity line until it intersects the diameter line, then follow the ordinate to the top of the page and find the friction head, 13 feet for 1000 foot run or 19.5 feet for the 1500 foot run. Second, let Q = 1.75 cubic feet per second and d = 8 inches. Trace to the left along the horizontal line representing the volume of 1.75 cubic feet until it intersects the diameter line, then read up and find the same friction head as before. Third, let the allowable friction head for 1500 feet of main be 19 feet, when Q = 1.75 cubic feet per second or when v = 5 feet per second. Reverse the process given above and find an 8 inch pipe.

Pipe Diameters and Friction Heads.*



*Church's "Hydraulic Motors."

Pipe Diameters and Friction Heads.*



*Church's "Hydraulic Motors."

TABLE 48.

Temperatures for Testing Direct Steam Radiation Plants.*

Gage pressure, lbs. per sq. in. Inches of vacuum.

Test condition	Steam Temperature	Steam pressure intended for zero weather										
		0 lb.	1 lb.	2 lb.	3 lb.	4 lb.	5 lb.	6 lb.	7 lb.	8 lb.	9 lb.	10 lb.
10 in.	192.0	63.3	62.3									
9 "	194.5	64.2	63.2	62.3								
8 "	197.0	65.0	64.0	63.0	62.2							
7 "	199.0	65.6	64.7	63.7	62.8	62.0						
6 "	201.0	66.3	65.3	64.3	63.4	62.6	62.0					
5 "	203.0	67.0	66.0	65.0	64.0	63.3	62.6	61.9				
4 "	205.0	67.6	66.6	65.6	64.7	63.9	63.2	62.5	61.7			
3 "	207.0	68.3	67.2	66.2	65.3	64.5	63.8	63.1	62.3	61.7		
2 "	208.5	68.8	67.7	66.7	65.7	65.0	64.2	63.6	62.8	62.0	61.5	
1 "	210.5	69.4	68.3	67.5	66.4	65.6	64.8	64.2	63.3	62.6	62.1	61.5
0 lb.	212.0	70.0	68.8	67.8	66.9	66.1	65.3	64.6	63.8	63.1	62.6	62.0
1 "	215.5	71.2	70.0	69.0	68.0	67.2	66.3	65.8	65.0	64.2	63.7	63.0
2 "	218.7	72.1	71.0	70.0	69.2	68.2	67.3	66.7	65.9	65.1	64.5	64.0
3 "	221.7		72.0	71.0	70.0	69.2	68.3	67.6	66.7	66.0	65.4	64.8
4 "	224.5			71.8	70.8	70.0	69.2	68.4	67.5	66.7	66.2	65.7
5 "	227.2				71.7	70.8	70.0	69.2	68.3	67.6	67.0	66.3
6 "	229.8					71.7	70.8	70.0	69.2	68.4	67.7	67.2
7 "	232.4						71.7	70.8	70.0	69.2	68.6	68.0
8 "	234.9							71.7	70.8	70.0	69.3	68.7
9 "	237.3								71.5	70.5	70.0	69.3
10 "	239.4									71.3	70.7	70.0
Factors		.670	.675	.678	.684	.688	.692	.694	.698	.702	.705	.707

The temperatures in this table are for a plant designed for 0° and 70°.

Example.—It is desired to test a plant designed for 5 pounds gage pressure on a day when the outside temperature is 22 degrees. What should be the temperature in the rooms with steam at 3 pounds gage pressure? It will be noted in the vertical column marked 5 pounds that opposite the 3 pound pressure 68.3 degrees may be expected on a zero day. As the temperature was 22 degrees above we must add 22 times .692, or 15.2 degrees, thus making a total of 83.5 degrees, the temperature which should exist indoors.

* W. W. Macon.

TABLE 49. **Specifications of Kewanee Firebox Boilers.*** (Steam and water heating.)

Number	0	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20
Capacity steam......sq. ft.	700	900	1000	1200	1400	1700	2100	2200	2500	2900	3200	4400	5800	8500	10500	12800
Capacity water......sq. ft.	1100	1400	1600	1900	2200	2700	3400	3500	4000	4600	5100	7000	9300	13600	16800	20500
Diameter ....inches	24	30	30	30	36	36	36	42	42	42	48	48	54	60	66	72
Length over all.........feet	7½	6½	7½	8½	7½	9	10½	8½	10	11½	10½	13½	16½	18	18	18
Width of firebox.....inches	19	24	24	24	30	30	30	36	36	36	42	42	48	54	60	66
Length of firebox....inches	26	26	32	38	32	38	44	38	44	50	44	56	62	68	68	74
Height of firebox....inches	30	36	36	36	41	41	41	43	43	43	47	47	49	54	59	64
Size of tubes..........inches	3	3	3	3	3	3	3	3	3	3	3	3	4	4	4	4
Length of tubes......inches	60	48	55	61	55	67	79	61	73	85	79	103	132	144	144	138
Square feet fire surface.....	93	120	138	154	186	222	258	259	304	348	393	502	647	954	1177	1442
Ratio rad. to fire surface...	7.4	7.5	7.3	7.8	7.5	7.7	8.2	8.5	8.2	8.3	8.2	8.8	8.9	8.9	8.9	8.9
Square feet of grate..........	3.30	4.3	5.0	6.33	6.33	8.0	8.8	9.5	11.0	12.5	12.5	16.0	20.6	25.3	28.3	33.8
Ratio fire surface to grate	28.0	28.0	27.0	24.0	29.0	27.0	29.0	27.0	28.0	28.0	31.0	31.0	31.0	38.0	42.0	43.0
Diam. of smoke pipe, inches	12	16	16	16	18	18	18	20	20	20	22	22	24	30	34	38
Size of steam supply, inches	2½	3	3	4	4	4	4	6	6	6	6	7	7	7	8	8
Size of return ....inches	2	2½	2½	3	3	3	3	4	4	4	4	5	5	5	6	6
Flow & ret., hot water...in.	2–3	2–4	2–4	2–4	2–5	2–5	2–5	2–5	2–6	2–6	2–6	2–7	2–7	2–8	2–10	2–10
Depth of water leg...inches	15	15	15	15	15	15	15	15	15	15	15	15	15	16	16	16
Shipping weight..........lbs.	2300	2450	2750	3150	3450	3850	4350	4500	5000	5650	6650	8000	10250	15250	18400	22000
Height of brickwork........	63″	69″	69″	69″	75″	75″	75″	81″	81″	81″	87″	87″	93″	103″	109″	115″
Height of water line........	48″	54″	54″	54″	59″	59″	59″	61″	61″	61″	65″	65″	67″	75″	80″	87″
Floor space{length	9′ 5″	8′ 8″	9′ 8″	10′ 8″	9′ 6″	11′ 2″	12′ 8″	11′ 1″	12′ 7″	14′ 1″	13′ 1″	16′ 1″	19′ 7″	21′ 1″	21′ 1″	21′ 5″
Floor space{width	4′ 6″	5′ 0″	5′ 0″	5′ 0″	5′ 6″	5′ 6″	5′ 6″	6′ 0″	6′ 0″	6′ 0″	6′ 6″	6′ 6″	7′ 8″	8′ 2″	8′ 8″	9′ 2″
Approximate number brick	1500	1600	1700	1800	2000	2300	2400	2500	2800	3000	3200	3700	5700	6500	7100	8000

Note.—A late modification of these boilers called the Kewanee "Smokeless" Firebox Boiler, with numbers increased by 100 and with capacities increased approximately 15 per cent. may be had in place of the above. The dimensions of the Smokeless boiler differ somewhat from those in this table. If exact dimensions are needed data must be obtained from the manufacturers

* Kewanee Catalog.

TABLE 50.

Percentage of Heat Transmitted by Various Pipe-Coverings, From Tests Made at Sibley College, Cornell University, and at Michigan University.*

Kind of covering	Relative amount of heat transmitted
Naked Pipe	100.
Two layers asbestos paper, 1 in. hair felt, and canvas cover	15.2
Two layers asbestos paper, 1 in. hair felt, canvas cover wrapped with manilla paper	15.
Two layers asbestos paper, 1 in. hair felt	17.
Hair felt sectional covering, asbestos lined	18.6
One thickness asbestos board	59.4
Four thicknesses asbestos paper	50.3
Two layers asbestos paper	77.7
Wool felt, asbestos lined	23.1
Wool felt with air spaces, asbestos lined	19.7
Wool felt, plaster paris lined	25.9
Asbestos molded, mixed with plaster paris	31.8
Asbestos felted, pure long fibre	20.1
Asbestos and sponge	18.8
Asbestos and wool felt	20.8
Magnesia, molded, applied in plastic condition	22.4
Magnesia, sectional	18.8
Mineral wool, sectional	19.3
Rock wool, fibrous	20.3
Rock wool, felted	20.9
Fossil meal, molded, ¾ inch thick	29.7
Pipe painted with black asphaltum	105.5
Pipe painted with light drab lead paint	108.7
Glossy white paint	95.0

*Carpenter's H. and V. B.

Note.—These tests agree remarkably well with a series made by Prof. M. E. Cooley of Michigan University, and also with some made by G. M. Brill, Syracuse, N. Y., and reported in Transactions of the American Society of Mechanical Engineers, Vol. XVI.

TABLE 51.

Factors of Evaporation.

Gage pressure	.3	10	20	30	50	100	125	135	150	175
Feed water	Factors of evaporation									
212	1.0003	1.0103	1.0169	1.0218	1.0290	1.0396	1.0431	1.0443	1.0460	1.0483
200	1.0127	1.0227	1.0293	1.0343	1.0414	1.0520	1.0555	1.0567	1.0584	1.0608
185	1.0282	1.0382	1.0448	1.0498	1.0569	1.0675	1.0710	1.0722	1.0739	1.0763
170	1.0437	1.0537	1.0603	1.0653	1.0724	1.0830	1.0865	1.0877	1.0894	1.0917
155	1.0592	1.0692	1.0758	1.0807	1.0878	1.0985	1.1020	1.1032	1.1048	1.1072
140	1.0715	1.0846	1.0912	1.0962	1.1033	1.1139	1.1174	1.1186	1.1203	1.1227
125	1.0901	1.1001	1.1067	1.1116	1.1187	1.1293	1.1328	1.1341	1.1357	1.1381
110	1.1055	1.1155	1.1221	1.1270	1.1341	1.1447	1.1482	1.1495	1.1511	1.1535
95	1.1209	1.1309	1.1375	1.1424	1.1495	1.1602	1.1637	1.1649	1.1665	1.1689
80	1.1363	1.1463	1.1529	1.1578	1.1650	1.1756	1.1791	1.1803	1.1820	1.1843
65	1.1517	1.1617	1.1683	1.1733	1.1804	1.1910	1.1945	1.1957	1.1974	1.1997
50	1.1672	1.1772	1.1838	1.1887	1.1958	1.2064	1.2099	1.2112	1.2128	1.2152
35	1.1827	1.1927	1.1993	1.2042	1.2113	1.2219	1.2255	1.2267	1.2283	1.2307

TABLE 52.

Per Cent. of Total Heat of Steam Saved per Degree Increase of Feed Water.

Initial temp. of feed	Gage pressure in boiler, lbs. per sq. in.									
	0	20	40	60	80	100	120	140	160	180
32	.0872	.0861	.0855	.0851	.0847	.0844	.0841	.0839	.0837	.0835
40	.0878	.0867	.0861	.0856	.0853	.0850	.0847	.0845	.0843	.0839
50	.0886	.0875	.0868	.0864	.0860	.0857	.0854	.0852	.0850	.0846
60	.0894	.0883	.0876	.0872	.0867	.0864	.0862	.0859	.0856	.0853
70	.0902	.0890	.0884	.0879	.0875	.0872	.0869	.0867	.0864	.0860
80	.0910	.0898	.0891	.0887	.0883	.0879	.0877	.0874	.0872	.0868
100	.0927	.0915	.0908	.0903	.0899	.0895	.0892	.0890	.0887	.0883
120	.0945	.0932	.0925	.0919	.0915	.0911	.0908	.0906	.0903	.0899
140	.0963	.0950	.0943	.0937	.0932	.0929	.0925	.0923	.0920	.0916
160	.0982	.0968	.0961	.0955	.0950	.0946	.0943	.0940	.0937	.0933
180	.1002	.0988	.0981	.0973	.0969	.0965	.0961	.0958	.0955	.0951
200	.1022	.1008	.0999	.0993	.0988	.0984	.0980	.0977	.0974	.0969
220	-------	.1029	.1019	.1013	.1008	.1004	.1000	.0997	.0994	.0989
240	-------	.1050	.1041	.1034	.1029	.1024	.1020	.1017	.1014	.1009

Example.—Boiler pressure 120 lbs. gage, initial temperature of feed water 60 deg., heated to 210 deg. Then increase in temperature 150, times tabular figure, .0862, equals 12.93 per cent. saving.

TABLE 53. **Vento Cast Iron Hot-Blast Heaters.**

		40″ Section						50″ Section						60″ Section					
		Width 9⅛″			Width 6¾″			Width 9⅛″			Width 6¾″			Width 9⅛″			Width 6¾″		
No. of loops in stack	† Width of standard stack in inches	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe	Square feet of heating surface	Net air space in square feet	Equivalent in lineal feet one-inch pipe
7	35	75.2	4.34	226	52.5	4.34	158	94.5	5.37	284	66.5	5.37	200	112.0	6.45	336	77.0	6.45	231
8	40	86.0	4.96	258	60.0	4.96	180	108.0	6.14	324	76.0	6.14	228	128.0	7.37	384	88.0	7.37	264
9	45	96.7	5.58	290	67.5	5.58	203	121.5	6.91	365	85.5	6.91	257	144.0	8.29	432	99.0	8.29	297
10	50	107.5	6.20	323	75.0	6.20	225	135.0	7.68	405	95.0	7.68	285	160.0	9.21	480	110.0	9.21	330
11	55	118.2	6.82	355	82.5	6.82	248	148.5	8.45	446	104.5	8.45	314	176.0	10.13	528	121.0	10.13	363
12	60	129.0	7.44	387	90.0	7.44	270	162.0	9.22	486	114.0	9.22	342	192.0	11.05	576	132.0	11.05	396
13	65	139.7	8.06	419	97.5	8.06	293	175.5	9.99	527	123.5	9.99	371	208.0	11.97	624	143.0	11.97	429
14	70	150.5	8.68	452	105.0	8.68	315	189.0	10.76	567	133.0	10.76	399	224.0	12.89	672	154.0	12.89	462
15	75	161.2	9.30	484	112.5	9.30	338	202.5	11.53	608	142.5	11.53	428	240.0	13.81	720	165.0	13.81	495
16	80	172.0	9.92	516	120.0	9.92	360	216.0	12.30	648	152.0	12.30	456	256.0	14.73	768	176.0	14.73	528
17	85	182.7	10.54	548	127.5	10.54	383	229.5	13.07	689	161.5	13.07	485	272.0	15.65	816	187.0	15.65	561
18	90	193.5	11.16	581	135.0	11.16	405	243.0	13.84	729	171.0	13.84	513	288.0	16.57	864	198.0	16.57	594
19	95	204.2	11.78	613	142.5	11.78	428	256.5	14.59	770	180.5	14.59	542	304.0	17.50	912	209.0	17.50	627
20	100	215.0	12.40	645	150.0	12.40	450	270.0	15.36	810	190.0	15.36	570	320.0	18.42	960	220.0	18.42	660
21	105	225.7	13.02	677	157.5	13.02	473	283.5	16.13	851	199.5	16.13	599	336.0	19.34	1008	231.0	19.34	693
22	110	236.5	13.64	710	165.0	13.64	495	297.0	16.90	891	209.0	16.90	627	352.0	20.26	1056	242.0	20.26	726
23	115	247.2	14.26	742	172.5	14.26	518	310.5	17.67	932	218.5	17.67	656	368.0	21.18	1104	253.0	21.18	759
24	120	258.0	14.88	774	180.0	14.88	540	324.0	18.44	972	228.0	18.44	684	384.0	22.10	1152	264.0	22.10	792

† Note.—Add to the width of stack 2½ inches for staggering of stacks. In addition to standard spacing (5″ centers), Ventos are also made having wide spacing (5⅜″ centers), and narrow spacing (4⅝″ centers). To change from standard to narrow spacing, multiply by .85, and from standard to wide, multiply by 1.18, for net air space.

Tapping 2½″ right hand on supply, 2½″ left hand on return, and bushed to required size. Steam and return on opposite ends, air vent on both ends. Steam and return on same end, air vent on same end.

TABLE 54.

Steam Consumption of Various Types of Non-Condensing Engines.* (Approximate).

Pounds per indicated horse-power hour.

Horse-power	Simple throt-tling 100 lbs. at throttle	Simple auto-matic 100 lbs. initial	Simple Corliss 100 lbs. initial	Simple four valve 100 lbs. initial	Com-pound four valve and Corliss 100 lbs. initial	Com-pound four valve and Corliss 125 lbs. initial	Com-pound four valve and Corliss 150 lbs. initial
10	52						
20	50	40.0					
30	49	39.0					
40	48	38.0					
50	48	38.0	34.5	35.0			
60	47	36.0	32.5	33.0			
70	47	35.0	31.5	32.0			
80	46	34.0	30.5	31.0			
90	45	33.0	29.5	30.0			
100	45	32.0	28.5	29.0			
150	44	31.5	28.0	28.5	22.5–23	21.5–22	21–21.5
200	43	30.5	27.0	27.5	22–22.5	21–21.5	20.5–21
250	43	30.0	26.5	27.0	22–22.5	21–21.5	20–20.5
300	42	29.0	25.5	26.0	22–22.5	20.5–21	20–20.5
400	41	28.5	25.0	25.5	21.5–22	20–20.5	19.5–20
500	41	28.5	25.0	25.5	20–21.5	19.5–20	19–19.5

The foregoing table was compiled principally from the records of a large number of actual tests of engines of various makes, under reasonably favorable conditions. It is based upon the actual weight of condensed exhaust steam.

* Atlas Engine Works Catalog.

TABLE 55.

Speeds, Capacities and Horse-Powers of "Green" Steel Plate Fans at Varying Pressures.*

Diam. wheel	Pressures	.26 in.	.87 in.	1.3 in.	1.7 in.	2.2 in.	2.6 in.	3.02 in.	3.46 in.	4.33 in.
		¼ oz.	½ oz.	¾ oz.	1 oz.	1¼ oz	1½ oz	1¾ oz.	2 oz.	2½ oz.
	CU. FT.	2249	3176	3891	4498	5029	5513	5956	6372	7135
30	R. P. M.	330	466	571	660	738	809	874	935	1047
	H. P.	.286	.811	1.491	2.298	3.213	4.227	5.311	6.515	9.120
	CU. FT.	3239	4581	5605	6477	7242	7937	8584	9173	10268
36	R. P. M.	275	389	476	550	615	674	729	779	872
	H. P.	.413	1.170	2.148	3.311	4.625	6.086	7.681	9.375	13.125
	CU. FT.	4398	6214	7617	8815	9864	10799	11679	12483	13981
42	R. P. M.	235	332	407	471	527	577	624	667	747
	H. P.	.557	1.576	2.898	5.473	6.300	8.287	10.450	12.750	17.825
	CU. FT.	5750	8123	9937	11500	12867	14123	15240	16301	18282
48	R. P. M.	206	291	356	412	461	506	546	584	655
	H. P.	.733	2.076	3.810	5.880	8.223	10.832	13.636	16.670	23.370
	CU. FT.	7602	10758	13167	15203	17030	18650	20145	21558	24174
54	R. P. M.	183	259	317	366	410	449	485	519	582
	H. P.	.970	2.750	5.047	7.767	10.880	14.300	18.017	21.992	30.896
	CU. FT.	9715	13718	16780	19429	21725	23786	25728	27495	30792
60	R. P. M.	165	233	285	330	369	404	437	467	523
	H. P.	1.241	3.506	6.433	9.932	13.882	18.230	22.996	28.077	39.355
	CU. FT.	12078	17071	20855	24156	26975	29551	32047	34221	38247
66	R. P. M.	150	212	259	300	335	367	398	425	475
	H. P.	1.542	4.361	7.996	12.352	17.238	22.666	28.675	35.123	48.895
	CU. FT.	15608	21942	26918	31103	34835	38115	41169	44109	49312
72	R. P. M.	138	194	238	275	308	337	364	390	436
	H. P.	1.983	5.601	10.322	15.881	22.252	29.223	36.808	45.043	62.783
	CU. FT.	20192	28405	34907	40383	45174	49452	53387	57152	63996
84	R. P. M.	118	166	204	236	264	289	312	334	374
	H. P.	2.581	7.262	13.387	20.650	28.875	37.931	47.775	58.450	81.812
	CU. FT.	23008	32614	39762	46016	51601	56515	60983	65227	73045
96	R. P. M.	103	146	178	206	231	253	273	292	327
	H. P.	2.941	8.337	15.261	23.531	32.982	43.348	54.511	66.707	93.380
	CU. FT.	29260	41027	50568	58519	65198	71559	77284	82690	92549
108	R. P. M.	92	129	159	184	205	225	243	260	291
	H. P.	3.737	10.488	19.397	30.060	41.666	54.871	69.163	84.556	118.291
	CU. FT.	36209	51042	62384	71982	80270	88559	95539	102083	114298
120	R. P. M.	83	117	143	165	184	203	219	234	262
	H. P.	4.628	13.050	23.925	36.807	51.307	67.928	85.495	104.401	146.116
	CU. FT.	43560	61565	75504	87120	97575	106868	115580	123711	138231
132	R. P. M.	75	106	130	150	168	184	199	213	238
	H. P.	5.568	15.730	28.957	44.550	62.370	82.096	103.430	126.521	176.715
	CU. FT.	52026	73138	89726	103298	116116	127426	137228	147030	164372
144	R. P. M.	69	97	119	137	154	169	182	195	218
	H. P.	6.65	18.700	34.411	52.822	74.221	97.741	122.802	150.371	210.133

Manufacturer's Note.—The horse-power required to drive a fan will vary according to the manner of application. The horse-powers given above are 25 per cent. greater than would be required under ideal conditions.

* Condensed from the G. F. E. Co. Catalog.

TABLE 56.

Speeds, Capacities and Horse-Powers of "A. B. C." Steel Plate Fans at Varying Pressures.*

Fan Number	Diam. of wheel	Static press.	½"	1"	1½"	2"	2½"	3"	3½"	4"
			.29 oz.	.58 oz.	.87 oz.	1.16 oz.	1.44 oz.	1.73 oz.	2.02 oz.	2.31 oz.
		C. F. M.	3840	5425	6640	7650	8595	9400	10110	10810
50	30	R. P. M.	471	665	816	945	1060	1150	1250	1330
		B. H. P.	.88	2.48	4.55	7.00	9.81	12.85	16.20	19.75
		C. F. M.	5475	7740	9460	10900	12250	13400	14410	15420
60	36	R. P. M.	393	555	681	786	880	961	1040	1110
		B. H. P.	1.25	3.53	6.49	9.94	14.00	18.35	23.10	28.10
		C. F. M.	7100	10020	12280	14150	15900	17400	18700	20010
70	42	R. P. M.	336	475	583	675	755	825	890	950
		B. H. P.	1.62	4.58	8.35	12.93	18.19	23.80	29.90	36.60
		C. F. M.	8640	12200	14950	17200	19350	21150	22800	24350
80	48	R. P. M.	294	416	511	590	660	722	780	832
		B. H. P.	1.97	5.57	10.20	15.71	22.10	28.90	36.50	44.50
		C. F. M.	11000	15540	19000	21900	24600	26950	29000	31000
90	54	R. P. M.	262	370	454	525	587	641	693	740
		B. H. P.	2.52	7.08	13.00	20.00	28.10	36.85	46.40	56.50
		C. F. M.	14050	19850	24300	28000	31450	34400	37000	39600
100	60	R. P. M.	236	333	409	473	529	578	625	665
		B. H. P.	3.21	9.05	16.65	25.60	35.95	47.10	59.10	72.30
		C. F. M.	16600	23500	28800	33100	37200	40700	43800	46900
110	66	R. P. M.	214	303	371	430	480	525	568	605
		B. H. P.	3.80	10.75	19.70	30.25	42.50	55.60	70.00	85.60
		C. F. M.	20300	28700	35100	40500	45500	49700	53500	57300
120	72	R. P. M.	196	278	340	394	440	481	520	555
		B. H. P.	4.64	13.10	24.00	37.00	52.00	68.00	85.50	104.50
		C. F. M.	27400	38700	47400	54500	61300	67000	72200	77250
140	84	R. P. M.	168	238	292	337	378	413	445	475
		B. H. P.	6.25	17.75	32.40	49.80	70.00	91.70	115.20	140.9
		C. F. M.	34500	48900	59800	68900	77300	84500	91000	97500
160	96	R. P. M.	147	208	256	296	331	362	390	416
		B. H. P.	7.88	22.30	41.00	62.90	88.40	115.5	145.4	178.0
		C. F. M.	42600	60300	73800	85000	95500	104300	112500	120000
180	108	R. P. M.	131	185	227	262	293	320	346	369
		B. H. P.	9.75	27.55	50.50	77.60	109.0	143.0	180.0	219.0
		C. F. M.	51600	73000	89400	103000	115700	126500	136100	145800
200	120	R. P. M.	118	166	204	236	264	289	312	332
		B. H. P.	11.8	33.30	61.20	93.50	132.1	173.0	217.50	266.0
		C. F. M.	61400	86800	106000	122200	137400	150200	162000	173000
220	132	R. P. M.	107	151	185	214	240	262	283	302
		B. H. P.	14.0	39.60	72.50	111.50	157.0	206.0	259.0	316.0
		C. F. M.	72000	101800	124500	143500	161000	176000	189500	203000
240	144	R. P. M.	98	139	170	197	220	241	260	377
		B. H. P.	16.5	46.50	85.00	131.00	184.0	241.0	303.0	370.5

Manufacturer's Note.—Any of the above fans, when running at the speed and pressure indicated, will deliver the volume of air and require no more power than given in the table.

Allowances must be made for the inefficiency of the motive power and for transmission losses between motive power and the fan.

* Condensed from the A. B. C. Co. Catalog.

TABLE 57.

Speeds, Capacities and Horse-Powers of "Sirocco" Fans at Varying Pressures.*

Fan number	Diam. of wheel	Pressures	¾ in.	1 in.	1¼ in.	1½ in.	2 in.	2½ in.	3 in.	3½ in.	4 in.
			.43 oz.	.58 oz.	.72 oz.	.87 oz.	1.16 oz.	1.44 oz.	1.73 oz.	2.02 oz.	2.31 oz.
4	24	C. F. M.	4260	4920	5500	6020	6945	7770	8520	9200	9840
		R. P. M.	391	453	505	554	640	714	783	846	905
		B. H. P.	.879	1.348	1.89	2.475	3.8	5.32	7.00	8.825	10.77
5	30	C. F. M.	6650	7690	8600	9416	10870	12150	13320	14380	15380
		R. P. M.	313	362	403	443	512	571	625	676	724
		B. H. P.	1.37	2.105	2.96	3.868	5.95	8.315	10.94	13.80	16.85
6	36	C. F. M.	9580	11060	12350	13540	15630	17470	19150	20680	22150
		R. P. M.	260	302	336	369	427	477	523	565	604
		B. H. P.	1.975	3.03	4.25	5.563	8.56	11.96	15.72	19.85	24.23
7	42	C. F. M.	13050	15070	16800	18425	21260	23800	26100	28200	30140
		R. P. M.	223	259	288	316	366	408	447	483	517
		B. H. P.	2.69	4.126	5.78	7.565	11.66	16.28	21.43	27.06	33
8	48	C. F. M.	17000	19700	22000	24100	27820	31100	34080	36800	39370
		R. P. M.	196	226	252	277	320	358	392	424	453
		B. H. P.	3.51	5.39	7.58	9.9	15.22	21.30	28.0	35.3	43.15
9	54	C. F. M.	21500	24860	27800	30440	35140	39300	43100	46600	49800
		R. P. M.	174	201	224	246	285	317	348	376	402
		B. H. P.	4.43	6.81	9.57	12.52	19.23	26.94	35.38	44.70	54.5
10	60	C. F. M.	26500	30750	34300	37650	43400	48570	53220	57500	61500
		R. P. M.	156	181	202	222	256	286	313	338	362
		B. H. P.	5.46	8.42	11.8	15.47	23.77	33.23	43.72	55.2	67.4
11	66	C. F. M.	32200	37200	41500	45530	52550	58830	64450	69630	74400
		R. P. M.	142	165	184	202	233	260	285	308	329
		B. H. P.	6.65	10.18	14.3	18.72	28.77	40.24	52.9	66.85	81.5
12	72	C. F. M.	38300	44240	49400	54130	62500	69900	76600	82800	88500
		R. P. M.	130	151	168	185	214	238	261	282	302
		B. H. P.	7.9	12.11	17	22.25	34.2	47.85	63	79.5	97
13	78	C. F. M.	45000	52000	58100	63600	73500	82100	90000	97300	104000
		R. P. M.	120	140	155	171	197	220	241	261	279
		B. H. P.	9.28	14.22	20	26.16	40.22	56.2	74	93.35	113.9
14	84	C. F. M.	52100	60200	67300	73700	85000	95000	104200	112700	120400
		R. P. M.	112	130	144	158	183	204	224	242	259
		B. H. P.	10.75	16.49	23.2	30.3	46.6	65	85.6	108	132
15	90	C. F. M.	59900	69230	77500	84700	97800	109200	119800	129600	138500
		R. P. M.	104	121	135	148	171	191	209	226	242
		B. H. P.	12.34	18.93	26.6	34.8	53.55	74.9	98.5	124.2	151.7
16	96	C. F. M.	67950	78430	81800	96140	114300	124500	136000	147000	157300
		R. P. M.	98	114	126	139	160	178	196	211	226
		B. H. P.	13.98	21.5	30.2	39.6	63	85.7	112	142	173

* Condensed from A. B. C. Co. Catalog.

APPENDIX II.

References used Chiefly in Refrigeration and Ice Production

TABLE 58.

Freezing Mixtures.*

Names and proportions of ingredients in parts	Reduction of temp. deg. F.		Total Reduction of temp. deg. F.
	From	To	
Snow or pounded ice 2; sodium chloride 1		— 5	
Snow 5; sodium chloride 2; ammonium chloride 1		—12	
Snow 12; sodium chloride 5; ammonium nitrate 5		—25	
Snow 8; calcium chloride 5	+32	—40	72
Snow 2; sodium chloride 1		— 5	
Snow 3; dilute sulphuric acid 2	+32	—23	55
Snow 3; hydrochloric acid 5	+32	—27	59
Snow 7; dilute nitric acid 4	+32	—30	62
Snow 3; potassium 4	+32	—51	83
Ammonium chloride 5; potassium nitrate 5; water 16	+50	+ 4	46
Ammonium nitrate 1; water 1	+50	+ 4	46
Ammonium chloride 5; potassium nitrate 5; sodium sulphate 8; water 16	+50	+ 4	46
Sodium sulphate 5; dil. sulphuric acid 4	+50	+ 3	47
Sodium nitrate 3; dil. nitric acid 2	+50	— 3	53
Amonium nitrate 1; sodium carbonate 1; water 1	+50	— 7	57
Sodium sulphate 6; ammonium chloride 4; potassium nitrate 2; dil. nitric acid 4	+50	—10	60
Sodium phosphate 9; dil. nitric acid 4	+50	—12	62
Sodium sulphate 6; ammonium nitrate 5; dil. nitric acid 4	+50	—14	64

TABLE 59.

Properties of Saturated Ammonia.†

Temp. deg. F.	Pressure absolute lbs. per sq. in.	Heat of vaporization	Vol. of vapor per lb. cu. ft.	Vol. of liquid per lb. cu. ft.	Wt. of vapor lbs. per cu. ft.
—40	10.69	579.67	24.38	.0234	.0411
—35	12.31	576.69	21.21	.0236	.0471
—30	14.13	573.69	18.67	.0237	.0535
—25	16.17	570.68	16.42	.0238	.0609
—20	18.45	567.67	14.48	.0240	.0690
—15	20.99	564.64	12.81	.0242	.0775
—10	23.77	561.61	11.36	.0243	.0880
— 5	27.57	558.56	9.89	.0244	.1011
+ 0	30.37	555.50	9.14	.0246	.1094
+ 5	34.17	552.43	8.04	.0247	.1243
+10	38.55	549.35	7.20	.0249	.1381
+20	47.95	543.15	5.82	.0252	.1721
+30	59.41	536.92	4.73	.0254	.2111
+40	73.00	530.63	3.88	.0257	.2577
+50	88.96	524.30	3.21	.02601	.3115
+60	107.60	517.93	2.67	.0265	.3745
+70	129.21	511.52	2.24	.0268	.4664
+80	154.11	504.66	1.89	.0272	.5291
+90	182.80	498.11	1.61	.0274	.6211
+100	215.14	491.50	1.36	.0279	.7353

*Tayler. Pocket Book of Refrigeration.
†Wood—Thermodynamics, Heat Motors and Refrigerating Machines.

TABLE 60.

Solubility of Ammonia in Water at Different Temperatures and Pressures. (Sims).*

1 lb. of water (also unit volume) absorbs the following quantities of ammonia.

Absolute pressure in lbs. per sq. in.	32° F.		68° F.		104° F.		212° F.	
	Lbs.	Vols.	Lbs.	Vols.	Lbs.	Vols.	Grms.	Vols.
14.67	0.899	1180	0.518	683	0.338	443	0.074	97
15.44	0.937	1231	0.535	703	0.349	458	0.078	102
16.41	0.980	1287	0.556	730	0.363	476	0.083	109
17.37	1.029	1351	0.574	754	0.378	496	0.088	115
18.34	1.077	1414	0.594	781	0.391	513	0.092	120
19.30	1.126	1478	0.613	805	0.404	531	0.096	126
20.27	1.177	1546	0.632	830	0.414	543	0.101	132
21.23	1.236	1615	0.651	855	0.425	558	0.106	139
22.19	1.283	1685	0.669	878	0.434	570	0.110	140
23.16	1.336	1754	0.685	894	0.445	584	0.115	151
24.13	1.388	1823	0.704	924	0.454	596	0.120	157
25.09	1.442	1894	0.722	948	0.463	609	0.125	164
26.06	1.496	1965	0.741	973	0.472	619	0.130	170
27.02	1.549	2034	0.761	999	0.479	629	0.135	177
27.99	1.603	2105	0.780	1023	0.486	638		
28.95	1.656	2175	0.801	1052	0.493	647		
30.88	1.758	2309	0.842	1106	0.511	671		
32.81	1.861	2444	0.881	1157	0.530	696		
34.74	1.966	2582	0.919	1207	0.547	718		
36.67	2.070	2718	0.955	1254	0.565	742		

TABLE 61.

Strength of Ammonia Liquor.*

Degrees Baume	Specific gravity	Percentage	Degrees Baume	Specific gravity	Percentage
10	1.0000	0.0	20	0.9333	17.4
11	0.9929	1.8	21	0.9271	19.4
12	0.9859	3.3	22	0.9210	21.4
13	0.9790	5.0	23	0.9150	23.4
14	0.9722	6.7	24	0.9090	25.3
15	0.9655	8.4	25	0.9032	27.7
16	0.9589	10.0	26 (a)	0.8974	30.1
17	0.9523	11.9	27	0.8917	32.5
18	0.9459	13.7	28	0.8860	35.2
19	0.9396	15.5	29	0.8805	

Note.—Sp. gr. of pure anhydrous ammonia = .623.
(a) Known to the trade as "29½ per cent."
*Tayler. Pocket-Book of Refrigeration.

TABLE 62.

Properties of Saturated Sulphur Dioxide. (Ledoux).*

Temp. of ebullition deg. F.	Absolute pressure lbs. per sq. in. P ÷ 144	Total heat from 32 deg. F.	Latent heat of vapor-ization	Heat of liquid from 32 deg. F.	Density of vapor wt. per cu. ft.
—22	5.56	157.43	176.99	—19.56	.076
—13	7.23	158.64	174.95	—16.30	.097
— 4	9.27	159.84	172.89	—13.05	.123
5	11.76	161.03	170.82	— 9.79	.153
14	14.74	162.20	168.73	— 6.53	.190
23	18.31	163.36	166.63	— 3.27	.232
32	22.53	164.51	164.51	0.00	.282
41	27.48	165.65	162.38	3.27	.340
50	33.25	166.78	160.23	6.55	.407
59	39.93	167.90	158.07	9.83	.483
68	47.61	168.99	155.89	13.11	.570
77	56.39	170.09	153.70	16.39	.669
86	66.36	171.17	151.49	19.69	.780
95	77.64	172.24	149.26	22.98	.906
104	90.31	173.30	147.02	26.28	1.046

TABLE 63.

Properties of Saturated Carbon Dioxide.†

Temp. of ebullition deg. F.	Absolute pressure in lbs. per sq. in.	Total heat from 32 deg. F.	Latent heat of vapor-ization	Heat of liquid from 32 deg. F.	Density of vapor or wt. per cu. ft.
—22	210	98.35	136.15	—37.80	2.321
—13	249	99.14	131.65	—32.51	2.759
— 4	292	99.88	126.79	—26.91	3.265
5	342	100.58	121.50	—20.92	3.853
14	396	101.21	115.70	—14.49	4.535
23	457	101.81	109.37	— 7.56	5.331
32	525	102.35	102.35	0.00	6.265
41	599	102.84	94.52	8.32	7.374
50	680	103.24	85.64	17.60	8.708
59	768	103.59	75.37	28.22	10.356
68	864	103.84	62.98	40.86	12.480
77	968	103.95	46.89	57.06	15.475
86	1080	103.72	19.28	84.44	21.519

* Kents' M. E. Pocket-Book.
† I. C. S. Pamphlet 1238 B.

TABLE 64.

Pressures and Boiling Points of Liquids Available for Use in Refrigerating Machines.*

Temperature of ebullition	Pressure of vapor Pounds per square inch absolute			
deg. F.	Sulphur dioxide	Ammonia	Carbon dioxide	Pictet fluid
—40		10.22		
—31		13.23		
—22	5.56	16.95		
—13	7.23	21.51	251.6	
— 4	9.27	27.04	292.9	13.5
5	11.76	33.67	340.1	16.2
14	14.75	41.58	393.4	19.3
23	18.31	50.91	453.4	22.9
32	22.53	61.85	520.4	26.9
41	27.48	74.55	594.8	31.2
50	33.26	89.21	676.9	36.2
59	39.93	105.99	766.9	41.7
68	47.62	125.08	864.9	48.1
77	56.39	146.64	971.1	55.6
86	66.37	170.83	1085.6	64.1
95	77.64	197.83	1207.9	73.2
104	90.32	227.76	1338.2	82.9

TABLE 65.

Table of Calcium Brine Solution.†

Deg. Baume 60 deg. F.	Per cent. calcium by weight	Lbs. per cu. ft. solution	Specific gravity	Specific heat	Freezing point deg. F.	Amm. gage pressure
0	0.000	0.0	1.000	1.000	32.00	47.31
2	1.886	2.5	1.014	.988	30.33	45.14
4	3.772	5.0	1.028	.972	28.58	43.00
6	5.658	7.5	1.043	.955	27.05	41.17
8	7.544	10.0	1.058	.936	25.52	39.35
10	9.430	12.5	1.074	.911	22.80	36.30
12	11.316	15.0	1.090	.890	19.70	32.93
14	13.202	17.5	1.107	.878	16.61	29.63
16	15.088	20.0	1.124	.866	13.67	27.04
18	16.974	22.5	1.142	.854	10.00	23.85
20	18.860	25.0	1.160	.844	4.60	19.43
22	20.746	27.5	1.179	.834	— 1.40	14.70
24	22.632	30.0	1.198	.817	— 8.60	9.96
26	24.518	32.5	1.218	.799	—17.10	5.22
28	26.404	35.0	1.239	.778	—27.00	.65
30	28.290	37.5	1.261	.757	—39.20	8.5″ vac.
32	30.176	40.0	1.283		—54.40	15″ vac.
34	32.062	42.5	1.306		—39.20	4″ vac.

* Kent's M. E. Pocket-Book.
† Am. Sch. of Cor. Dickerman-Boyer.

TABLE 66.

Table of Salt Brine Solution.*

(Sodium chloride).

Degrees Salometer at 60 deg. F.	Per cent. by wt. of salt	Pounds of salt per cu. ft.	Specific gravity	Specific heat	Freezing point deg. F.	Amm. gage pressure
0	0.00	0.000	1.0000	1.000	32.0	47.32
5	1.25	0.785	1.0090	.990	30.3	45.10
10	2.50	1.586	1.0181	.980	28.6	43.03
15	3.75	2.401	1.0271	.970	26.9	41.00
20	5.00	3.239	1.0362	.960	25.2	38.96
25	6.25	4.099	1.0455	.943	23.6	37.19
30	7.50	4.967	1.0547	.926	22.0	35.44
35	8.75	5.834	1.0640	.909	20.4	33.69
40	10.00	6.709	1.0733	.892	18.7	31.93
45	11.25	7.622	1.0828	.883	17.1	30.33
50	12.50	8.542	1.0923	.874	15.5	28.73
55	13.75	9.462	1.1018	.864	13.9	27.24
60	15.00	10.389	1.1114	.855	12.2	25.76
65	16.25	11.384	1.1213	.848	10.7	24.46
70	17.50	12.387	1.1312	.842	9.2	23.16
75	18.75	13.396	1.1411	.835	7.7	21.82
80	20.00	14.421	1.1511	.829	6.1	20.43
85	21.25	15.461	1.1614	.818	4.6	19.16
90	22.50	16.508	1.1717	.806	3.1	18.20
95	23.75	17.555	1.1820	.795	1.6	16.88
100	25.00	18.610	1.1923	.783	0.0	15.67

TABLE 67.

Horse-Power Required to Produce One Ton of Refrigeration.†

Condenser pressure and temperature.

Refrigerator press. and temp.	P	103	115	127	139	153	168	184	200	218
P	T	65	70	75	80	85	90	95	100	105
4	—20°	1.0584	1.1304	1.2051	1.2832	1.3611	1.4427	1.5251	1.6090	1.6910
6	—15	.9972	1.0694	1.1450	1.2221	1.3001	1.4101	1.4609	1.5458	1.6300
9	—10	.9026	.9777	1.0453	1.1183	1.1926	1.2602	1.3471	1.4352	1.5093
13	— 5	.8184	.8833	.9537	1.0230	1.0935	1.1679	1.2437	1.3209	1.3961
16	0	.7352	.8008	.8648	.9328	1.0019	1.0718	1.1467	1.2194	1.2547
20	5	.6665	.7312	.7946	.8593	.9278	.9978	1.0656	1.1381	1.2121
24	10	.5915	.6629	.7257	.7894	.8545	.9205	.9911	1.0595	1.1294
28	15	.5410	.5998	.6641	.7276	.7924	.8553	.9224	.9943	1.0603
33	20	.4745	.5340	.5923	.6716	.7148	.7796	.8420	.9031	.9736
39	25	.4103	.4659	.5227	.5804	.5992	.7022	.7667	.8289	.8922
45	30	.3509	.4056	.4612	.5178	.5755	.6353	.6944	.7590	.8172
51	35	.3005	.3546	.4101	.4666	.5214	.5804	.6398	.7009	.7629

Note.—The above figures are purely theoretical. In practice about 50 per cent. must be added.

* Am. Sch. of Cor. Dickerman-Boyer.

† De La Vergne Catalog.

TABLE 68.

Cubic Feet of Ammonia Gas per Minute to Produce One Ton of Refrigeration per Day.*

Condenser pressure and temperature.

Refrigerator pressure and temperature	Press.	103	115	127	139	153	168	185	200	218
Press.	Temp.	65°	70°	75°	80°	85°	90°	95°	100°	105°
4	—20°	5.84	5.90	5.96	6.03	6.06	6.16	6.23	6.30	6.43
6	—15°	5.35	5.40	5.46	5.52	5.58	5.64	5.70	5.77	5.83
9	—10°	4.66	4.73	4.76	4.81	4.86	4.91	4.97	5.05	5.08
13	— 5°	4.09	4.12	4.17	4.21	4.25	4.30	4.35	4.40	4.44
16	0°	3.59	3.63	3.66	3.70	3.74	3.78	3.83	3.87	3.91
20	5°	3.20	3.24	3.27	3.30	3.34	3.38	3.41	3.45	3.49
24	10°	2.87	2.90	2.93	2.96	2.99	3.02	3.06	3.09	3.12
28	15°	2.59	2.61	2.65	2.68	2.71	2.73	2.76	2.80	2.82
33	20°	2.31	2.34	2.36	2.38	2.41	2.44	2.46	2.49	2.51
39	25°	2.06	2.08	2.10	2.12	2.15	2.17	2.20	2.22	2.24
45	30°	1.85	1.87	1.89	1.91	1.93	1.95	1.97	2.00	2.01
51	35°	1.70	1.72	1.74	1.76	1.77	1.79	1.81	1.83	1.85

TABLE 69.

Table of Refrigerating Capacities.†

Size of building				Number of cu. ft. per ton of refrigeration at temperatures given						
				Temperatures						
Dimensions of building	Contents cu. ft.	Surface in sq. ft.	Ratio cu. ft. to sq. ft.	0°	8°	16°	24°	32°	40°	48°
5x4x5	100	130	1.3	900	1100	1300	1500	1700	1900	2100
8x10x10	800	520	.65	1800	2200	2600	3000	3400	3800	4200
25x40x10	10000	3300	.33	3600	4400	5200	6000	6700	7600	8400
20x50x20	20000	4800	.24	4860	5940	7020	8100	9180	10260	11340
40x50x20	40000	7600	.19	6300	7700	9100	10500	11900	13300	14700
60x50x20	60000	10400	.17	6840	8360	9880	11400	12920	14440	15960
80x50x20	80000	13200	.165	7200	8800	10700	12000	13600	15200	16800
100x50x20	100000	16000	.16	7200	8800	10400	12000	13600	15200	16800
100x100x20	200000	28000	.14	8100	9900	11700	13000	15300	17100	18900
100x100x40	400000	36000	.09	13050	15950	18850	21750	24650	27550	30450
100x100x60	600000	44000	.073	16200	19800	23400	27000	30600	34200	37800
100x100x80	800000	52000	.065	18000	22000	26000	30000	34000	38000	42000
100x100x100	1000000	60000	.06	19350	23650	27950	32250	36550	40850	45150

* Featherstone Foundry and Machine Co. Catalog.
† Tayler. P. B. of R.

TABLE 70.

Approximate Cost of Ice Making.*

Tons ice per day	Engineers $2.50 to $5.00 per day	Oilers $2.00 per day	Firemen $1.50 to $1.75 per day	Tankmen and laborers $1.25 to $1.50 per day	Coal $2.00 per ton	Oil, waste, light and sundries	Daily operating expenses	Cost of ice per ton
10	2 at $4.50			2 at $3.00	3600 at $3.60	$1.50	$12.60	$1.26
20	2 " 5.00		2 at $3.00	2 " 3.00	6600 " 6.60	2.00	19.60	.98
25	2 " 5.25		2 " 3.00	2 " 3.00	8000 " 8.00	2.50	21.75	.87
30	2 " 5.50		2 " 3.00	2 " 3.00	9300 " 9.30	3.00	23.80	.79
40	2 " 6.00		2 " 3.00	3 " 4.50	12300 " 12.30	3.50	29.30	.76
60	3 " 9.00	1 at $2.00	3 " 4.50	3 " 4.50	18000 " 18.00	4.00	42.00	.70
75	3 " 10.00	1 " 2.00	3 " 4.50	4 " 6.00	22000 " 22.00	4.50	49.00	.65⅓
100	3 " 11.00	1 " 2.00	4 " 6.00	6 " 9.00	28500 " 28.50	5.00	61.50	.61½
120	3 " 11.50	1 " 2.00	4 " 6.00	6 " 9.00	34000 " 34.00	5.50	67.50	.56¼

This table does not include such charges as delivery, interest, taxes, etc.

* Featherstone Foundry and Machine Co.

TABLE 71.

Temperatures to Which Ammonia Gas Is Raised by Compression.*

Temperature of suction	Absolute condensing pressure	Absolute suction pressure					
		20	25	30	35	40	45
0 deg. F.	90	199	165	138	116	98	83
	110	232	196	166	145	126	109
	130	261	222	193	169	150	132
	150	285	246	216	191	171	153
	160	296	257	226	202	181	163
5 deg. F.	90	266	172	145	123	104	89
	110	239	203	174	151	132	115
	130	268	230	200	176	156	139
	150	293	254	223	198	178	160
	160	305	265	234	209	188	170
10 deg. F.	90	213	178	151	129	110	96
	110	247	210	181	158	139	122
	130	275	237	207	183	163	145
	150	301	262	231	205	185	167
	160	313	273	241	216	195	176
15 deg. F.	90	221	185	158	135	117	101
	110	254	217	188	164	145	128
	130	283	245	214	191	170	152
	150	309	269	238	213	192	173
	160	321	281	249	223	202	183
20 deg. F.	90	228	192	164	141	123	106
	110	262	224	195	171	150	134
	130	291	252	222	197	176	158
	150	317	277	245	220	198	180
	160	329	288	256	230	209	190
25 deg. F.	90	235	199	171	148	129	111
	110	269	230	200	178	155	140
	130	299	259	229	204	183	165
	150	325	284	253	227	205	187
	160	338	296	264	237	216	197
30 deg. F.	90	242	206	177	154	134	118
	110	277	239	208	184	164	147
	130	307	267	236	211	190	171
	150	334	292	260	234	212	193
	160	346	304	271	245	223	203
35 deg. F.	90	249	213	182	160	141	124
	110	286	246	215	191	170	153
	130	315	274	243	217	196	178
	150	341	300	268	241	219	200
	160	354	312	279	252	230	210

*Tayler. P. B. of R.

TABLE 72. **Comparison of Various Hydrometer Scales. (Yaryan).***

Degrees Baume	0	5	10	15	20	25	30	35	40	45	50	55	60	65	70
Degrees Densimetric 15.5° C. (60° F.)	0	3.6	7.4	11.5	16.0	20.8	26.1	31.8	38.1	45.0	52.6	61.1	70.6	81.2	93.3
Degrees Twaddell 60° F. T° = 200 (sp. gr. —1)	0	7.2	14.8	23.0	32.0	41.6	52.2	63.6	76.2	90.0	105.2	122.2	141.2	162.4	186.6
Degrees Brix. 15.5° C. Official Prussian. $\text{sp. gr.} = \frac{400}{400 - \text{B X}^\circ}$	0	13.9	27.5	41.3	55.2	68.9	82.8	96.5	110.3	124.1	137.9	151.7	165.5	179.3	193.0
Degrees Beck 12.5° C. $\text{sp. gr.} = \frac{170}{170 - \text{B K}^\circ}$	0	5.9	11.7	17.6	23.5	29.3	35.2	41.0	46.9	52.8	58.6	64.5	70.4	76.2	82.1
Degrees Brix. Saccharimetric. (per cent. sugar)	0	9.0	18.0	27.0	36.2	45.5	55.1	64.7	74.7	85.1	------	------	------	------	------
Gay-Lussac (C) $\text{sp. gr.} = \frac{100}{100 - \text{C}^\circ}$	0	3.5	6.9	10.3	13.8	17.2	20.7	24.1	27.6	31.0	34.5	37.9	41.4	44.8	48.3
Specific gravity — U. S. Chem. Asso. 15.5° C. (60° F.) — Liquids heavier than water $\text{sp. gr.} = \frac{145}{145 - \text{B}^\circ}$	1	1.036	1.074	1.115	1.160	1.208	1.261	1.318	1.381	1.450	1.526	1.611	1.706	1.812	1.933
Specific gravity — U. S. Chem. Asso. 15.5° C. (60° F.) — Liquids lighter than water $\text{sp. gr.} = \frac{140}{130 + \text{B}^\circ}$	---	------	1.000	0.966	0.933	0.903	0.875	0.849	0.824	0.800	0.778	0.757	0.737	0.718	0.700
Specific gravity — Modulus 144.38. Custom in France	1	1.0380	1.0745	1.1160	1.1607	1.2095	1.2625	1.3200	1.3830	1.4525	1.5300	1.6150	1.7110	1.8185	1.9410

*Tayler. P. B. of R.

TABLE 73.

Time Required to Freeze Ice in Cells or Cans. (a) (Siebert).*

Temp. deg. F.	Thickness in inches											
	1	2	3	4	5	6	7	8	9	10	11	12
10	0.32	1.28	2.86	5.10	8.00	11.5	15.6	20.4	25.8	31.8	38.5	45.8
12	0.35	1.40	3.15	5.60	8.75	12.6	17.3	22.4	28.4	35.0	42.3	50.4
14	0.39	1.56	3.50	6.22	9.70	14.0	19.0	25.0	31.5	39.0	47.0	56.0
16	0.44	1.75	3.94	7.00	11.00	15.8	21.5	28.0	35.5	43.7	53.0	63.0
18	0.50	2.00	4.50	8.00	12.50	18.0	24.5	32.0	40.5	50.0	60.5	72.0
20	0.58	2.32	5.25	9.30	14.60	21.0	28.5	37.3	47.2	58.3	70.5	84.0
22	0.70	2.80	6.30	11.20	17.50	25.2	34.3	44.8	56.7	70.0	84.7	100.0
24	0.88	3.50	7.86	14.00	21.00	31.5	42.8	56.0	71.0	87.5	106.0	126.0

(a) Time required from one wall, for plate ice, two times the above values.

TABLE 74.

Standard Sizes of Ice Cans.†

Size of cake, in pounds	Size of top, inches	Size of bottom, inches	Inside depth, inches	Outside depth, inches	Size of band, inches
50	8x8	7½x7½	31	32	¼x1½
100	8x16	7¼x15¼	31	32	¼x1½
200	11½x22½	10½x21½	31	32	¼x2
300	11½x22½	10½x21½	44	45	¼x2
400	11½x22½	10½x21½	57	58	¼x2

TABLE 75.

Cold Storage Temperatures for Various Articles.*

Article	Temp. deg. F.	Article	Temp. deg. F.	Article	Temp. deg. F.
Apples	32–36	Fruits	26–55	Oranges	45–50
Asparagus	34	Fruits (dried)	35–40	Oysters	33–35
Bananas	40–45	Fruits (canned)	35	Oysters (in tubs)	25
Beans (dried)	32–40	Furs (undressed)	35	Oysters (in shells)	33
Berries (fresh)	36–40	Furs (dressed)	25–32	Peaches	45–55
Buckwheat flour	40	Game (frozen)	25–28	Pears	34–36
Butter	32–38	Game (to freeze)	15–28	Peas (dried)	40
Cabbage	34	Grapes	36–38	Pork	34
Cantaloupes	40	Hams	30–35	Potatoes	36–40
Celery	32–34	Hops	33–40	Poultry (frozen)	28–30
Cheese	32–33	Honey	45	Poultry (to freeze)	18–22
Chocolate	40	Lard	34–45	Sugar, etc.	40–45
Cider	30–40	Lemons	36–40	Syrup	35
Claret	45–50	Meat (canned)	35	Tobacco	35
Corn (dried)	35	Meat (fresh)	34	Tomatoes	36
Cranberries	34–36	Meat (frozen)	25–28	Vegetables	34–40
Cream	35	Milk	32	Watermelons	34
Cucumbers	39	Nuts	35	Wheat flour	40
Dates	55	Oat meal	40	Wines	40–45
Eggs	33–35	Oil	35	Woollens, etc.	25–32
Figs	55	Oleomargarine	35		
Fish (fresh)	25–30	Onions	34–40		
Fish (dried)	35				

* Tayler. P. B. of R.

† As adopted by the Ice Machine Builders' Association of the U. S.

APPENDIX III.

Tests of House Heating Boilers.

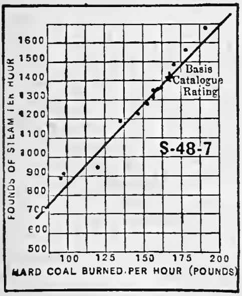
The following extract from a series of tests on a Number S-48-7 Ideal Sectional Boiler from the reports of the American Radiator Company's Institute of Thermal Research, Buffalo, New York, will be of interest.

Size of Grate.............48x64½ in. Grate area............. 21.6 sq. ft.
Heating surface—total ....300.0 sq. ft.

0—Fuel used in tests	Hard Coal	Hard Coal	Hard Coal
1—No. of boiler	S-48-7	S-48-7	S-48-7
2—Duration of test hours	8:00	7:00	8:00
4—Fuel burned during test, lbs.	1360.00	1344.00	1434.00
5—Fuel per hour, lbs.	170.00	192.00	178.20
6—Fuel per sq. ft. grate per hour, lbs.	7.90	8.95	8.35
7—Stack temperature, degrees Fahrenheit	750.00	725.00	600.00
8—Evaporation per sq. ft. of heating surface per hour, lbs.	4.97	5.60	5.24
9—Evaporative power available—lbs. of water per lb. of coal	8.80	8.75	8.77
10—Boiler-power (evaporation per hour)—lbs. (item 5 × item 9)	1496.00	1680.00	1562.00
11—Capacity—sq. ft. (item 10 ÷ 0.22)	6800.00	7640.00	7100.00
12—Capacity—sq. ft. (item 10 ÷ 0.25)	5980.00	6720.00	6250.00

Catalog rating ....5700 sq. ft.

The accompanying figure shows the combustion chart as developed for this same boiler. The tests were run to find the *evaporative power* and *capacity* with varying amounts of coal burned per hour. Coal was fired at regular intervals and the steam pressure was maintained at two pounds gage on the radiation. Line 11 gives the capacity in square feet of radiation including mains and risers, at the rate of .22 pound of steam per square foot per hour. Line 12 gives the capacity at .25 pound of steam per square foot per hour. In average service about one-third of these quantities of coal would be burned. The catalog rating is based upon burning 167.5 pounds of coal per hour and an evaporation of 8.5 pounds of water per pound of coal (rates of combustion and evaporation that seem justifiable). As will be seen from lines 5 and 9 the actual amount of coal



burned and the actual evaporation in each test exceed this figure. Multiplying 167.5 by the assumed evaporative rate of 8.5 and dividing by .25 = 5700 square feet. Comparing with column 2, line 5 times line 9 divided by .25 gives 6720 square feet, which is above the catalog rating. Test number two compared with test number one shows that by increasing the amount of coal from 170 pounds to 192 pounds per hour increases the boiler capacity 740 square feet.

Data Required for Estimating Plain Hot Water or Steam Plants.

Name of room	Location exposed or not	Size of room			Cubic contents	Sq. ft. exp. glass	Sq. ft. exp. wall	Radiators Steam or water						Remarks: Cold floor, ceiling, etc.
		Long	Wide	High				Direct	Direct indirect	Indirect	Number	Style	High	

Date..192...

Owner of building........................Address........................

Architect ....Address........................

Kind of building..........................Location..........................

Nearest freight station...

Temperature in living rooms............ Kind of fuel used..................

Height of cellar................ Size of smoke flue.......... × ....in.

Items to Estimate on.

Boiler and foundation ....

Smoke pipe and damper ....

Thermometers and pressure and safety gages.................................

Draft regulation ....

Firing tools ....

Filling and blow-off connection ....

Pipe and fittings ....

Sq. ft. of radiation --

Cut-off valves and radiator valves --

Air valves --

Radiator wall shields --

Temperature control --

Humidifying apparatus ---

Floor and ceiling plates ---

Hangers ---

Expansion tank --

Cold air ducts, stack boxes and registers---

Pipe covering --

Bronzing ---

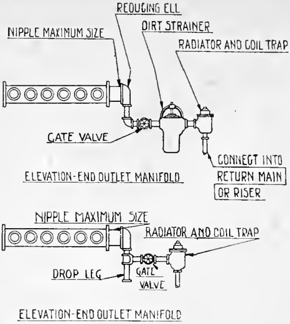
Labor of installation ---

Freight and cartage --

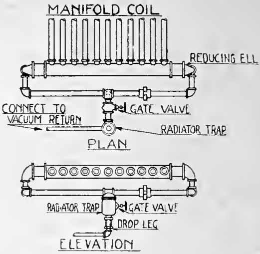
Per cent. of profit ---

Total bid ---

Submitted by --



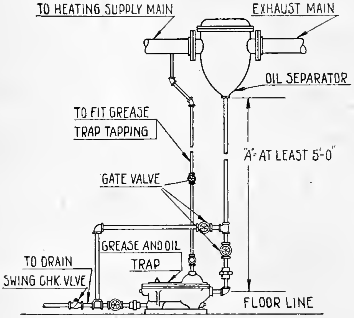
A



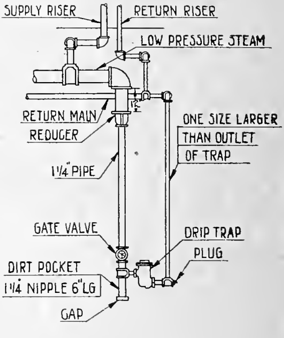
B

Method of installing return connections from overhead manifold coils, using drop legs, traps and dirt strainers.

A for 6 coils or less, B for more than 6 coils.



C

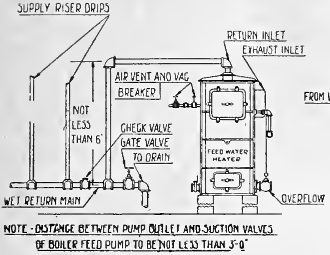


D

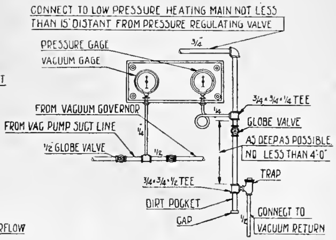
C. Method of installing drip connections from horizontal oil separator through grease and oil trap to drain.

D. Method of dripping steam supply main through drip trap into vacuum return, using vertical loop as cooling surface and dirt pocket.

*Warren Webster Co.



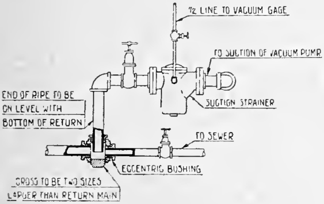
E



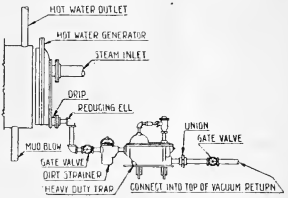
F

E. Method of draining down feed supply risers through wet return into a feed water heater.

F. Method of making connections to gages.



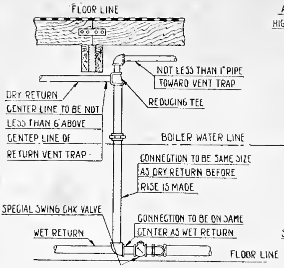
G



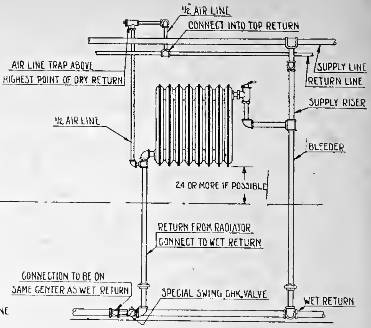
H

G. Method of installing a suction strainer where return main rises to vacuum pump, using fittings for lift pocket.

H. Method of draining a hot water generator through a gate valve, dirt strainer and heavy duty trap.



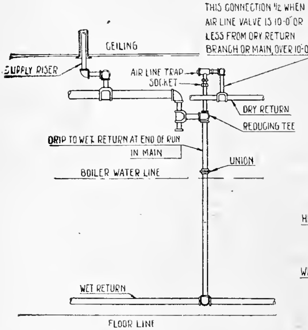
I

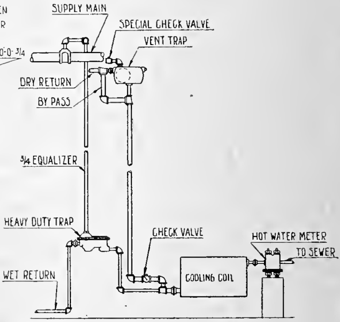


J

I. Method of installing connections where dry return rises and drips into wet returns.

J. Detail showing return connection from radiators on brackets to wet return near floor with air line connections through air line trap into dry return near ceiling.





K L

K. Method of installing drip connection at end of supply main, or end of a long supply main branch.

L. Arrangement of return for modulation system where steam is taken from outside source.

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